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MARINE ENGINEERING

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MARINE ENGINEERING

(A TEXT-BOOK)

BY

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ROYAL NAVY (RETIRED)

LATE INSTRUCTOR IN STEAM AND MARINE ENGINEERING, MACHINE CONSTRUCTION, ETC.
AT THE ROYAL NAVAL COLLEGE, GREENWICH
AND LECTURER AT THE ROYAL NAVAL WAR COLLEGE
AUTHOR OF 'TURBINES,' ETC.
MEMBER OF THE INSTITUTION OF MECHANICAL ENGINEERS, ETC.

FIFTH EDITION

ENTIRELY REVISED, WITH ABOUT 480 ILLUSTRATIONS

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TO

MY WIFE .

MY BEST FRIEND AND STERNEST CRITIC

520655

PREFACE TO FIFTH EDITION

The Fourth Edition was published a few weeks before the outbreak of the Great War, and owing to active service it was impossible to start on this revision before the middle of 1919. The enormous progress in Marine Engineering during the last five years has necessitated the entire rewriting of many chapters and a considerable revision of the remainder. The opportunity has been taken to cut out as much obsolete or obsolescent matter as possible to make room for a fuller consideration of mercantile practice, particularly relating to geared turbines and their auxiliaries.

The general arrangement of the book is based on the path of the steam from its generation to its condensation and return to the boiler as feed water. The section on Turbines has been enlarged and divided into three chapters which now follow the section on the Reciprocating Engine. The latest systems of Oil Fuel Firing are illustrated and described. The section on Internal Combustion Engines has been enlarged, and the subject, including submarine and mercantile engines, is treated as fully as the near future is likely to demand.

The book is intended to cover the syllabus of a sound marine engineering course for all those who are interested in the subject, and when it is considered that at least one volume could be written on the matter treated in each chapter, making at least thirty-eight volumes, it may be appreciated that the compilation of a single volume of this nature is no small task.

The first three years of the war having been spent in supervising repairs to transports and H.M. ships by numerous private firms in the Clyde district, and the remaining time on similar duties in

N.W. Italy, I have added considerably to my knowledge and experience, which now numbers over forty-one years.

I have availed myself of much information from the Engineering Press, particularly *Engineering*, *The Engineer*, *The Times Engineering Supplement*, and *International Marine Engineering*, and the *Proceedings of the Institution of Mechanical Engineers*. Many firms have supplied me with information, details, illustrations, and descriptions of their specialities. The sources of information are acknowledged in the letterpress, and I wish to express my sincere thanks to every one concerned.

Materials for independent judgment are usually given, and conclusions are drawn only when in strict accord with known facts of the best practice of the present day. Engineering is in a continuous process of evolution, and it is necessary to bring an unbiassed mind to bear on every problem which comes before us, and throughout I have tried to keep this object in view.

My best thanks are given to a brother-officer for his careful reading and constructive criticism of this revision.

A. E. TOMPKINS.

Brockenhurst,
December 1920.

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PART I

CHAPTER I

THE MOTIVE POWER OF THE ENGINE, AND ITS DEVELOPMENT

Introductory. — In the ordinary steam engine, a force or pressure of steam acts on one side of a piston, or circular disc, and thus produces motion of the piston. This force overcomes another force opposed to it, which may be considered to act on the opposite side of the piston, and motion thus produced is applied to some useful purpose in doing work.

As shown in Fig. 1, suppose such a piston can be moved easily by hand, without allowing steam to leak past its edge, inside a cylinder which has its bottom end closed. Let the piston rest on a little water in the bottom of the cylinder, and then apply a fire so as to heat the water. At first the piston is pushed upwards as the water gets hotter, because the water expands slightly, but the distance the piston moves is not easily measured. If the piston be very light, and practically weightless, the water will begin to boil at a temperature of 212° Fahr.; and, steam being formed, the piston is pushed upwards because the steam thus produced fills 1643 times the volume or space that the water does from which it is formed. Thus, if all the water be changed into steam, the piston rises to a height 1643 times the original depth of the water.

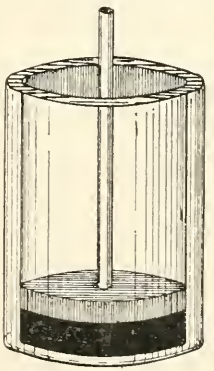


Fig. 1.—Piston and Cylinder.

If the piston be very heavy, or loaded with a heavy weight, the water does not begin to boil until it is much hotter than before, and

the steam also fills less space or volume. Consequently, the piston rises to only a moderate height, as shown in Fig. 2; therefore it is seen that, when steam is formed, a large weight can be lifted, and this simply by heating water beneath a piston in a cylinder.

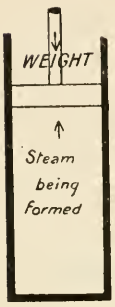


Fig. 2.

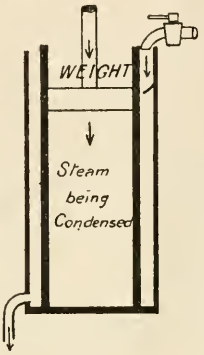


Fig. 3.

Next, suppose the piston and weight have been lifted, and that the fire is now removed, and that the cylinder is cooled by spraying cold water round it (as shown in Fig. 3); then the steam is condensed, and, filling less space or volume as water instead of steam, the piston falls back to its original position.

In an engine as above described, three separate operations are performed in the cylinder. Steam is generated, and the cylinder thus acts as a *boiler*; the piston is moved, and work is done in overcoming resistance, thus making the cylinder and piston a machine, or *engine*; and the steam is condensed to water, thus the cylinder acts as a *condenser*.

In 1698 Savery took out a patent "for raising water by means of the elastic force of steam." This engine was apparently never usefully employed, although, in a modified form, its action is very similar to that of the well-known pulsometer pump.

In 1712 Newcomen, who appears to have worked in conjunction with Savery, produced the first commercially useful engine, and it was erected at Dudley Castle. For the first time a separate vessel (*B* in Fig. 4) was used, and steam was generated in it as a *boiler*.

The figure shows the engine in its position of rest when not at work; the piston is at the top of its stroke in the cylinder *F*. On top of the standard *XX* is a crossbeam, *LKM*, pivoted at *K*. To the end *M* the piston, working in the cylinder *F*, is suspended by chains. To the other end *L* the plunger *P* and pump rod are similarly suspended; and to these is added a weight, *W*, so that there is a preponderance at this end *L*. Chocks of wood, *NN*, limit the vertical see-saw movement of the crossbeam about its pivot at *K*, and therefore also limit the *stroke*, or up-and-down movement, of the steam piston and pump plunger.

The fire is lighted under the boiler *B*, on the firegrate *C*. Steam,

being generated in the boiler, is admitted at first by hand, but afterwards automatically, to the cylinder *F*, through the cock *A*. After completing the first downward stroke of the piston, the greater weight at the end *L* causes the piston to rise, and allows the steam to fill the space below the piston in the cylinder. When the piston has moved through about two-thirds of the upward stroke, the cock *A* is closed,

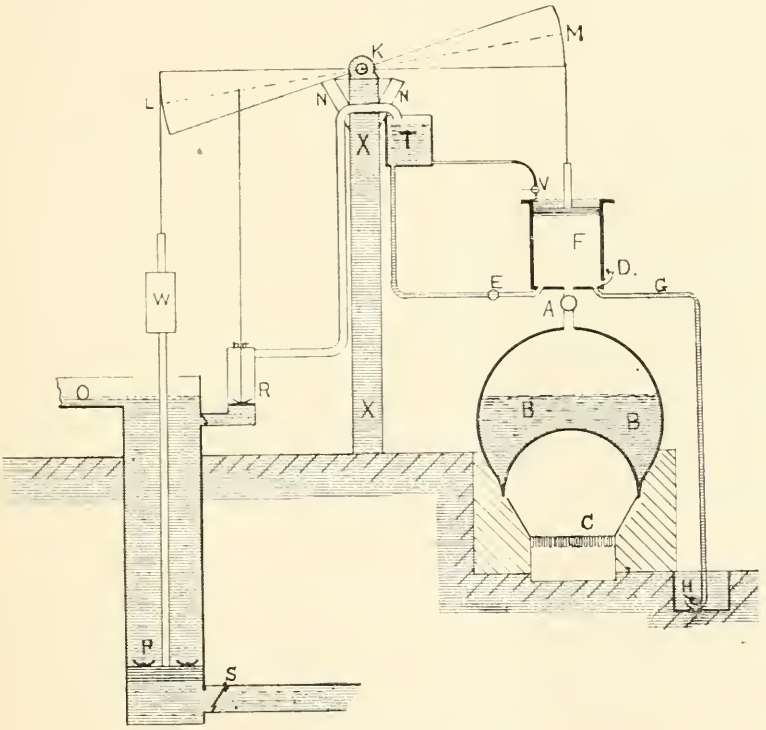


FIG. 4.—Newcomen's Pumping Engine.

and the steam inside the cylinder expands to fill the greater volume given it by further movement. Just before the end of the upward stroke, the injection cock *E* is opened, admitting cold water into "the heart of the steam," which is almost instantly condensed to water. The mixed condensed steam and water falls to the bottom of the cylinder, leaving the upper space, below the piston, practically clear of pressure of vapour (in other words, an almost complete *vacuum* is formed). The atmospheric pressure, or weight of the atmosphere, presses on top of the piston and forces it downward, due to a difference

of pressure of about 11 lb. (14·7-3·7) per square inch of piston area. This force, pulling the end *M* of the beam downwards, raises the weight and the pump bucket with the water above it, and discharges some water through the opening at *O*. Just before the end of the downward stroke, the steam is again admitted to the cylinder, and, cushioning the piston, brings it gently, and without shock, to rest at the end of the stroke ; the water is expelled from the cylinder into a hot-well, or tank, *H*, through a pipe *G* and non-return valve.

The smaller pump *R* simply serves to keep the tank *T* supplied with water for injection purposes, and for sealing the piston to prevent air-leakage past its edge into the cylinder. An escape valve, *D*, weighted to about 2 lb. per square inch above the pressure of the atmosphere, prevents any undue pressure accumulating in the cylinder, and also lets out any excess of the injection water.

The workmanship of this engine was necessarily extremely rude ; the engine derived its power principally from the atmospheric pressure on top of the piston, and was *single-acting*—that is, the power was obtained on the down-stroke only, and the preponderance of weight at *L* produced the upward movement. It was soon found that if too much injection water was used, although a much lower pressure was obtained by more rapid condensation, a large quantity of steam was wasted in reheating the cylinder at each upward stroke ; the fuel expenditure increased when the temperature of the outgoing eduction water was reduced below about 150° F. The range in temperature of the steam in the cylinder was about 212°-150°, or 62° F., which corresponds very closely with that adopted for economical working in the H.P. and M.P. cylinders of to-day. Some of these engines had steam cylinders, made of brass, 6 feet in diameter and 9 feet piston stroke.

About 1765 James Watt began to make improvements in the steam engine by introducing a separate vessel, called a *condenser*, in which the steam could be condensed (*H* in Fig. 5). The object of this is to keep the cylinder warm, so that the steam on first entering from the boiler should not be so freely used and condensed in heating the cylinder to its own temperature, and therefore be able to act more promptly in raising the piston. At the same time, it is necessary to keep the condenser continually pumped clear of the condensed steam and water used for cooling it. The cooling water weighs roughly 40 to 60 times the weight of the steam which it is used to condense ; and, in addition, a large volume of air and other uncondensable gases enters the cylinder and condenser with the steam and water, and as this volume is very large compared with the water, the pump used for the purpose of clearing the condenser is called the *air pump*, and is shown to the left of the condenser at *N*.

In a further endeavour to keep the cylinder warm, and to prevent

cold air from entering it above the piston as in Newcomen's engine, Watt encased the cylinder in another and outer cylinder, as shown in the figure. In so doing he closed the top of the cylinder, except where the piston rod *R* passes through it. At this place a stuffing box and gland is fitted around the rod, so that neither can steam escape from the cylinder nor can air leak into it. The engine now became *double-acting*, and independent of the atmospheric pressure, for, by an arrangement of taps or valves, steam could be admitted above the

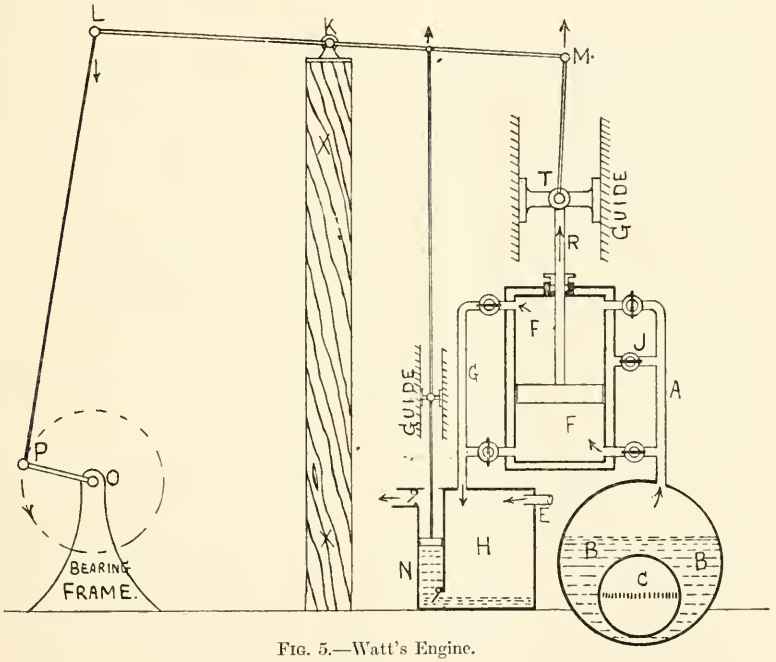


FIG. 5.—Watt's Engine.

piston to force it downwards as well as upwards. This arrangement of taps is shown in the figure, with each one opened or closed as necessary for the upward stroke of the piston; steam then enters through *A* from the boiler, below the piston, and the escaping or exhaust steam above the piston passes away through the pipe *G* to the condenser *H*.

With this arrangement the *steam jacket* was also used with the same object, that of keeping the cylinder warm, by allowing steam to enter by a pipe and valve, *J*, into the *jacket space* between the inner and outer cylinders. (In some cases it was arranged for the steam to pass through the jacket on its way to the cylinder; this was

a common but wasteful method, and is now seldom used.) The water condensed from the jacket steam, in its endeavour to keep the cylinder warm, is taken away by a drain pipe (not shown) to the condenser or open air.

The top end of the piston rod was connected with the end of the see-saw beam *LKM* by a *connecting rod TM*, with pin joints at each end. To keep the piston rod in line with the centre of the piston and cylinder, and to prevent its being bent by oblique forces, a *crosshead*, *T*, was also fitted and guided by two parallel bars, called the guides, or *guide*.

About 1783 Watt applied his engine to rotate a shaft, *O*, at first through a sun-and-planet motion, but ultimately through a connecting rod, *LP*, and a crank, *OP*. An engine of this type was used by John Fitch of Connecticut, about 1786, for moving a skiff by paddles.

With only one cylinder, and when the piston is on the top or bottom of its stroke, called the dead points or *dead centres*, it is evident that the engine can exert no force to rotate the crank, and therefore a *flywheel* was fitted on the shaft. The heavy flywheel, gathering energy of rotation during the working parts of the strokes, gives out some of its energy at the dead centres by continuing its motion, and thus tends to promote a uniform speed of rotation of the shaft. (The flywheel is seldom fitted to marine engines, but it is a common feature of petrol and other small engines.)

Later an improvement was made which, to a large extent, rendered the flywheel unnecessary. By making two cylinders to each engine, and coupling their pistons respectively with two cranks side by side and at right angles to each other on the same shaft, one piston is always in a position to produce motion. This arrangement is very convenient for starting and reversing heavy engines.

In all the very early engines, the admission of steam to, and the opening of the escape or exhaust from, the cylinders were controlled by hand. Later the valves were worked by the engine itself, so that after the engine was once started it continued working automatically, and was, therefore, self-contained. Still later, the slide valve was introduced, by which both admission and exhaust are controlled by a single valve. (The slide valve is fully explained later, in its proper place.)

In 1801 Symington constructed the first *direct-acting engine*, in the form which is now most commonly used, thus dispensing with the beam, as shown in Fig. 6. The piston rod is here shown connected

directly with the crank through the medium of the connecting rod. By turning the picture into various positions, we get—a *horizontal engine*, when the piston moves to and fro in a horizontal line ; a *vertical engine*, when the crank is at the top and the cylinder at the bottom ; a *vertical inverted engine*, commonly called the vertical engine, and generally used for marine purposes, when the cylinder is at the top and the crank at the bottom ; or an *inclined engine*, when the piston works to and fro in some line other than vertical or horizontal.

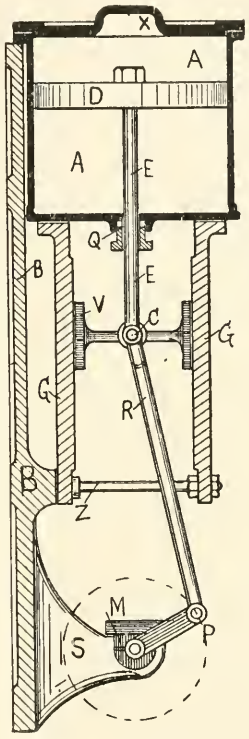


FIG. 6.—Direct-acting Engine.

A, Cylinder.
B, Frame.
C, Crosshead.
D, Piston.
E, Piston rod.
G, Guide.
M Main bearing (for shaft).
O, Centre of shaft.
P, Crank.
Q, Packing gland.
R, Connecting rod.
S, Part of frame *B* (and holding *M*).
V, Slipper (part of *C*).
X, Cylinder cover.
Z, Stay between guides.

Historical Notes.—About 1781 Hornblower suggested taking the exhaust steam from an engine exhausting to the atmosphere and utilising it in another cylinder. This system of *compound*, or stage-expansion, *engines* did not come into general use until about 1860. In 1782 Watt improved the movement of the admission and exhaust valves so as to cut off the supply of steam to the cylinder before the piston finished its stroke, thus using the steam expansively. *Expansive working* was a feature of all the early engines, but was not entirely understood. Owing to the difficulty of reversing, the early locomotives were nearly all practically non-expansive working, and it was not until about 1840 that *lap* was added to the *slide valve*, and an increased economy of about 30 per cent was obtained by the consequent increased work done during expansion.

In 1787–88 Symington and Taylor built a twin boat propelled by *paddle wheels* worked by a steam engine of about one horse-power, which obtained a speed of five miles an hour. From this early effort

grew the extensive use of paddle-wheel propelled ships, and it is noteworthy that a steam-paddle frigate took part in the bombardment of Acre in 1840.

In 1785 the *screw propeller* was proposed by Watt, and patented by Bramah for the propulsion of vessels, but was first practically used by Stevens (U.S.A.) in a small boat driven by a double-acting engine. Stevens also invented *closed ashpit draught*, a system which is now becoming extensively used in the Navy, although very common in the mercantile marine for some thirty years past. It was only, however, in 1836 that serious efforts were made, by Petit Smith, with screws. He succeeded in attaining about $9\frac{1}{4}$ knots[1] with the *Archimedes*. This favourable result led to the introduction of screw propellers in H.M.S. *Mermaid* and *Rattler*.

The early marine engines worked at a pressure, by gauge, of about 4 or 5 lb. per square inch ; and in 1860 the pressure had risen to about 20 or 30 lb. per square inch. *Jet condensers* were then usual ; but the lime scale formed in the boilers, and left on the heating surfaces, from the use of sea water, prevented an increase of pressure with safety. The introduction of the *surface condenser* about 1860, by which the boilers received fresh or nearly fresh feed water, allowed an increase of pressure to about 35 lb. Next an improvement in the shape of boiler, from the square box to the cylindrical type, allowed a further increase to 60 lb., and compound (two-stage expansion) engines also came into fashion. About 1887 the introduction of the *evaporator* for making sea water into fresh water, the general improvement in manufacture of boilers and machinery, the use of *mild steel* with a higher allowable working stress, and the *corrugated furnace*, allowed a further increase in pressure to about 155 lb., and three-stage (and later four-stage) expansion engines became usual. In 1893 the introduction of water-tube boilers allowed a further increase of pressure, which gradually rose to about 300 lb. in 1897. Beyond this no further increase has been made for boilers, but the pressure for engines has risen to about 270 lb.

Types of engines now merely of historical interest, but still to be found in old vessels, are the oscillating engine invented by Murdock in 1790, trunk engine (a form now used *for petrol and oil-gas* engines), and the return connecting-rod engine. The horizontal direct-acting engine generally took the place of the above for naval purposes, but was soon

[1] 1 nautical mile = 6080 feet. 1 knot = 1 nautical mile, or sea mile, per hour Thus $9\frac{1}{4}$ sea miles per hour = 10·65 land miles per hour = $9\frac{1}{4}$ knots.

superseded by the vertical inverted engine, which in its turn is giving way to turbines.

Stage Expansion.—If there are two or more cylinders to one engine (which is commonly the case) and the steam passes into each cylinder *directly* from the boiler, and is exhausted from each cylinder *directly* into the atmosphere or condenser, the engine is a *simple* or *single-stage expansion* engine. A diagram of the course of the steam for two cylinders is shown in Fig. 7.

If the steam passes into one cylinder from the boiler, and then passes into another cylinder, as shown in Fig. 8—the steam is evidently expanded in two stages and does work successively in two cylinders,—the engine is called a *two-stage expansion* engine, which is also frequently called a compound engine ; but the term *compound* is also applicable

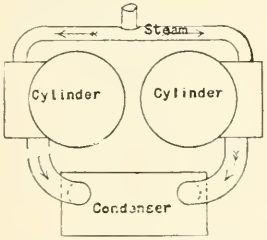


FIG. 7.—Single-stage Expansion Two-cylinder Engine.

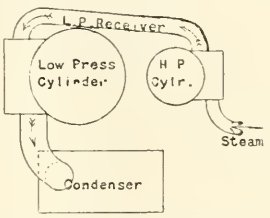


FIG. 8.—Two-stage Expansion Two-cylinder Engine.

to all engines with two or more stages of expansion. (The vertical inverted cylinder compound engine was introduced by John Elder in 1854.)

If the steam passes through three cylinders in succession, the engine is called a *three-stage* or triple expansion engine. Similarly for four cylinders in succession, the engine is called a *four-stage* or quadruple expansion engine. And so on, for five or more cylinders.

The course of the steam through a three-stage expansion engine with three cylinders is shown in Fig. 9. The *receivers* are simply pipes and spaces necessary for connecting successive cylinders ; these spaces form a *reservoir* (a term which has been corrupted into *receiver*) from which each cylinder receives its supply of steam. From each cylinder the steam is exhausted into the next lower stage reservoir (or receiver), and in the final stage it is exhausted into the condenser through the eduction pipe.

Cycle of Operations.—The three primary elements of a marine

steam plant are the boiler, the engine or mechanism, and the condenser. The cycle of operations is started by raising steam from water in the *boiler*; this steam is conveyed away, through the boiler stop valve and the *steam pipe*, to the engine-regulating valve and the *engine*; in the engine the steam does work, by virtue of its heat energy, and is exhausted to the condenser; in the condenser the steam not already condensed during its passage through the engine is condensed to water by the cooling action of the injection water, or of the *circulating water* in a *surface condenser*; from the condenser the condensed water is

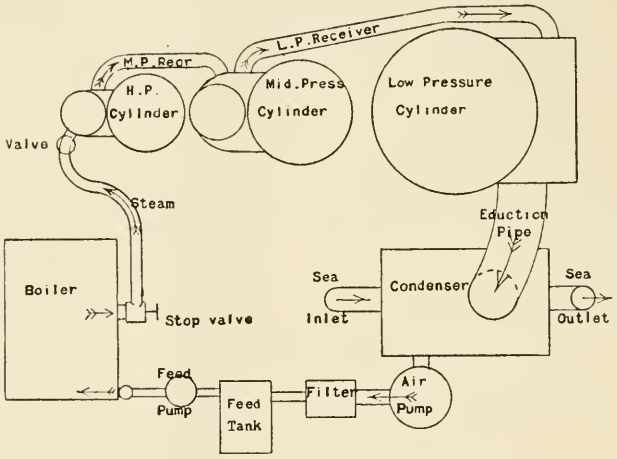


Fig. 9.—Three-stage Expansion Engine and Cycle of Operations.

pumped by the *air pump* into the *feed tank*, and the air, except a moderate quantity which remains in solution in the water, is allowed to escape; finally, the *feed pump* takes the water from the feed tank and delivers it into the boiler, where another similar cycle of operations begins.

In some cases the water is heated, before being returned to the boiler, by a *feed heater* or an economiser. Some portion of the water is lost by leakage during the cycle of operations, and to "make up" this loss an *evaporator* is used to convert sea water into fresh-water vapour or steam; this vapour is conducted to the condenser, and is condensed with the engine exhaust steam.

Fig. 9A shows the course of the steam through a three-stage expansion turbine set of marine engines fitted with Single Reduction Gearing. The H.P. and M.P. turbines are fitted on one shaft, and the

L.P. on a separate shaft. The two turbine shafts are geared into a single set of double helical gearing, and drive one propeller shaft. Two or more sets of such engines may be fitted in one ship according to the number of shafts and propellers.

The above summary gives all the principal parts of an ordinary

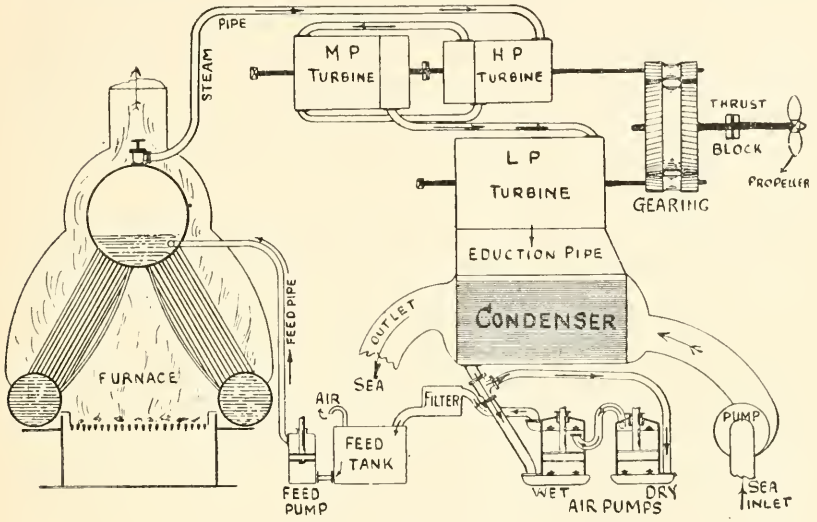


FIG. 9A.—Three-stage Turbine Engine and Cycle of Operations.

and simple steam plant; but there are also a number of other useful and essential fittings which are described in detail in other chapters, with the elements to which they belong.

CHAPTER II

UNITS—FORCE, WORK, POWER, ENERGY, PRESSURE, ETC.

In all machines, of which the steam engine is one, a force is exerted at one part which overcomes a resistance at another part; motion is produced and work is done. For the measurements of these forces and resistances some definite units are necessary for comparing the relative values in each machine, and also for comparing one machine with another. These units are given below, and cover all the ordinary requirements of marine and mechanical engineering. A short explanation is also given where it is considered necessary.

Length (L, *l*).—The standard of length is *one yard*, but in engineering the unit is one-third of a yard, or *one foot.*

Time (*t*).—The unit of time is *one second.*

Velocity (V or *v*).—The unit of velocity is *one foot per second.*

The motion of a body is *uniform* when it passes through equal spaces in equal times; it is *variable* when it passes through unequal spaces in equal times, or through equal spaces in unequal times. If the motion is an increasing one, it is *accelerated*; but if it is a decreasing one, it is generally termed *retarded* or *decelerated.*

For the proper measurement of velocity, the *time interval* must be very small. Thus a train may cover the distance between equally spaced stations in equal times, and therefore with an equal *average* velocity from station to station; but the velocity is not uniform, because the actual velocity of the train, from instant to instant, varies considerably.

Volume (V or *v*).—A bounded part of matter is called a body; and the part of space which a body occupies is called its *volume.*

The British unit of capacity, or volume, is *one gallon*, which contains exactly 10 lb. weight, at sea-level at Greenwich, of *pure water* at the standard temperature of 40° F.

Mass (M or m).—The quantity of matter contained in a body is called its *mass*. The unit of mass is a constant quantity, and does not change with its position relatively to the earth's surface.

Weight (W or w).—The unit of weight is *one pound*, which is equal to the weight of a standard mass, or quantity of matter, *in vacuo*.

The weight of a body, although it may be considered constant for engineering purposes, changes with its distance from the centre of the earth. The acceleration of velocity due to gravity (g) is found by careful experiment to vary, in exactly the same proportion as the weight, when both are measured at the same positions relative to the centre of the earth. The mass of a body is always equal to its weight divided by the gravitational acceleration ; or—

$$m = \frac{w}{g}.$$

At sea-level at Greenwich, the acceleration due to gravity is approximately equal to 32·16 feet per second in each second, and therefore, for ordinary engineering purposes—

$$m = \frac{\text{weight of the same body}}{32{\cdot}2}, \text{ approximately.}$$

Force.—A body which is at rest (*i.e.* remaining at the same place in space) cannot change its position without a cause ; nor can a moving body change the conditions of its motion without a cause. Such a cause is called *force*.

The engineering unit of force is *one pound*, and is the force that can balance the weight of 1 lb. at sea-level at Greenwich.

An *effort* is said to be applied when a force is caused to act tending to produce motion of a body. A *resistance* is a force which acts opposite to the direction of, and is called into play by, an effort.

Force also changes the " state of rest " of a body ; that is, it may call an increased frictional resistance into play without either reaching the limiting friction or just producing motion.

If a force of 1 lb. produces motion of a body, it produces a certain rate of change, or acceleration, in its velocity. The unit of force may also be defined as that force which can give unit acceleration to a body of unit mass. It is evident that this unit of force is a constant quantity ; because the unit of acceleration, being 1 foot per second in each second, does not vary ; and the unit of mass, being also a constant quantity, does not vary. This definition of force is in agreement with the one previously given, and is merely another way of defining the same quantity.

Density.—The density of a substance is the mass contained in one unit of volume of the substance. The density varies with the temperature and the pressure to which a substance is subjected.

All substances are compressible; but the practicable compression of many solid and liquid substances is so small that the change in density may usually be neglected.

Specific Density, or Gravity.—The relative density, or specific gravity, of a substance is the ratio of the density of a substance to that of *pure water* at a standard temperature. This temperature is often taken at 39° Fahr., which is the temperature of water at about its maximum density. The relative density of a gas is usually taken as the ratio of the density of the gas to that of *air* at 32° Fahr., and at a pressure of one atmosphere (14·7 lb. per square inch). (*Note.*—In chemistry and physical experiments and data, *hydrogen* instead of *air* is used for the standard of comparison.)

Specific Volume.—The specific volume, or bulkiness, of a substance is the number of units of volume occupied by unit mass of the substance.

For steam, it is measured by the number of cubic feet to 1 lb. by weight at any given pressure.

Relative Volume.—This is the ratio of the volume of the steam produced to that of the water from which it is generated.

One pint of water, weighing 1¼ lb., when completely evaporated into steam, fills 1643 pints under normal atmospheric pressure. If this steam be compressed under a pressure of about 21 atmospheres, the *relative volume* becomes about 90 pints. If the pressure be decreased, so as to become about two-fifteenths the normal atmospheric pressure (corresponding to a vacuum of 26 inches on the mercury barometer), this steam then fills about 11,000 pints, and its relative volume is about 11,000. (See column 3 of table, p. 18.)

Work.—Work is said to be done when motion is produced against resistance. The product of the resistance in terms of units of force, and the distance in terms of units of length, through which the resistance is overcome, is therefore a measure of the work done against the resistance.

The unit of work is *one foot-pound* (1 foot-lb.).

Other units are sometimes used in engineering which are self-explanatory, such as foot-tons, inch-tons, etc.

Power.—The power of an engine is the rate at which it can do work, *i.e.* the quantity of work it can do in a certain time.

The unit of power is *one foot-pound-second*; but as this unit is small, it is more convenient to use a derived unit of power—this is *one horse-power* (H.P.), which is equal to 33,000 *foot-lb.* of work per *minute.*

One *cheval vapeur* (French H.P.) is equal to 32,552 foot-lb. per minute, or 75 kilogrammetres per *second*.

Energy.—The capacity of bodies for doing work is called energy, and a body is said to contain more energy the more work it can do, regardless of time. The terms *energy* and *power* must not be considered to mean the same; power has relation to time. Thus 1 ton of coal may, by being burnt in a sufficiently large boiler, make sufficient steam to produce 1200 horse-power for 1 hour; whereas, when burnt in a small boiler, it may produce 120 horse-power for 10 hours—the energy contained in the coal being the same in both cases.

Energy appears in various forms — mechanical energy, heat, electrical energy, etc.; but they are all mutually convertible, as we shall see later.

Pressure.—When a force is acting it is generally distributed over an area (even when the force is acting on a point or line, because both a point and line may be considered to have some area); the unit of area in engineering is taken either as 1 square foot or as 1 *square inch*. For pistons, plungers, cylinders, etc., used in fluid machines, the unit is commonly taken as 1 square inch.

The part of the force exerted on each square inch is called the pressure (p), or pressure per square inch (or the intensity of pressure per square inch, or unital pressure), and it is evident that, if the pressure is uniform, the pressure on each square inch being equal, this pressure p, multiplied by the area A in square inches, is equal to the force exerted, F; or—

$$p \times A = F.$$

Atmospheric Pressure.—The pressure of the atmosphere depends on the relative density of the atmosphere, and at Greenwich, corrected to sea-level, is 14·7 lb. per square inch when the mercury barometer stands at 30 inches. In other words, when the atmospheric pressure is 14·7 lb. per square inch, it will support a column of mercury 30 inches in height, 762 mm.

The pressure of the atmosphere is actually equal to the weight of a vertical column of the atmosphere, whose height is equal to the height of the atmosphere at the place of measurement and whose horizontal area is the unit of area on which the pressure is exerted.

Absolute Pressure.—Absolute pressure (p) is measured from the *zero pressure* of the atmosphere; it is therefore =(gauge pressure +14·7) lb. per square inch.

The nought (0), or zero, shown on pressure, combined pressure and vacuum, and vacuum gauges, always corresponds to atmospheric pressure (14·7 lb. per square inch). Thus if a boiler pressure-gauge shows 60, it means 60 lb. *above* atmospheric, or 74·7 lb. *absolute* pressure. When the vacuum gauge shows 30 inches, it means there is zero absolute pressure, or 14·7 lb. *below* atmospheric. For purposes of general application, an *increase* of vacuum of 2 inches represents a *decreased* pressure, below atmospheric, of 1 lb. per square inch.

Pressure of 1 kilogramme per square centimetre = 14·223 lb. per square inch. 1 lb. per square inch = 0·0703 kilogrammes per square centimetre.

Mechanical Energy.—There are two kinds of mechanical energy. One is *potential*, or that of position, and is best illustrated by that of a weight suspended at some point above the earth. The other is *kinetic*, and is representative of any body in motion, and may be illustrated by a rotating flywheel. Both the weight and the flywheel possess capacity to do work, or energy: the former in falling to some lower level, and the latter by reason of its motion.

When an engine is rotating a shaft or performing other work, with a uniform rate of motion and overcoming a uniform resistance, then—

Energy developed = work done (in the same units of measurement);

but if the motion be accelerated while the resistance remains uniform, then—

$$\text{Energy developed} = \text{work done} + \left\{\begin{array}{l}\text{kinetic energy accumulated in}\\ \text{the mechanism.}\end{array}\right.$$

The acceleration may be either positive or negative, the velocity being respectively either increased or decreased. If the acceleration be negative, then work is done at the expense of the kinetic energy previously accumulated in the mechanism, which is correspondingly reduced, and—

$$\text{Energy developed} = \text{work done} - \left\{\begin{array}{l}\text{kinetic energy taken from the}\\ \text{mechanism.}\end{array}\right.$$

As an illustration, suppose a weight, W, is lifted through a height, H, then the work done = W × H in foot-lb. The weight then possesses potential energy of W × H foot-lb. due to its position. As the weight falls it loses potential energy and accumulates energy of motion, or kinetic energy. In falling through any distance H, the velocity V accumulated by the falling body is found by many concise experiments to be (where g = acceleration due to gravity = 32·2 feet per second per second)—

$$V = \sqrt{2g} \cdot H,$$

or $$V^2 = 2g \cdot H,$$

and therefore $$H = \frac{V^2}{2g}.$$

Multiply both sides of this equation by W ; then—

$$W \cdot H = \frac{W \cdot V^2}{2g}.$$

Potential energy = W . H = $\frac{1}{2}m$. V^2 = kinetic energy.

If the weight falls through only a part, h, of its height, H, then at this point we obtain the equation (where v is the velocity accumulated at the end of the fall h)—

$$W \cdot H = W(H - h) + W \cdot \frac{v^2}{2g}.$$

Or,

Potential energy originally available = potential energy not utilised + kinetic energy accumulated.

Although, for all ordinary purposes, the old steam tables are accurate enough, because the difference in values seldom exceeds one-half per cent, it appears better to use the new tables, derived from Dr. Callender's experiments for turbine calculations, especially in connection with nozzles. The values in the Table of Properties of Saturated Steam, given on the next page, are extracted from "The New Steam Tables," by Messrs. Smith and Warren. It will be noticed that the Latent Heat of 1 lb. of steam at atmospheric pressure is now given as 970·5 b.t.u. instead of the original 966.

1 calorie = 3·968 b.t.u. 1 b.t.u. = 0·252 calories.
1 calorie per kilogramme = 1·8 b.t.u. per 1 lb.
1 b.t.u. per 1 lb. = 0·554 calories per kilogramme.

[TABLE C

Properties of Saturated Steam (Pound-Fahrenheit Units).

p	t	$\frac{v}{s}$	w	v	S	L	L+S=H	p_0
Gauge Pressure per sq. inch.	Boiling-point Fahr.	Relative Volume.	Weight of 1 cub. ft.	Volume of 1 lb. Weight.	Sensible Heat from 32° Fahr.	Latent Heat.	Total Heat of Evaporation of 1 lb. Steam from 32° Fahr.	Absolute Pressure per sq. inch.
Lb.	Degrees		Lb.	Cub. ft.	B. Th. Units.	B. Th. Units.	B. Th. units.	Lb.
Inches of mercury. 28·99	79·9	40,000	0·00154	640·3	47·9	1044·9	1092·8	0·5
27·97	102·0	20,810	0·00300	333	70·1	1032·9	1103·0	1·0
25·9	126·3	10,845	0·00576	173·4	94·5	1019·7	1114·2	2·0
23·87	141·6	7,656	0·00843	118·5	109·9	1011·4	1121·2	3·0
21·83	152·9	5,654	0·01105	90·47	121·2	1005·0	1126·3	4·0
19·79	162·3	4,587	0·01362	73·39	130·7	999·8	1130·5	5·0
17·75	170·1	3,870	0·01616	61·87	138·6	995·2	1133·8	6·0
15·71	176·9	3,347	0·01866	53·56	145·4	991·3	1136·7	7·0
13·67	182·9	2,954	0·02115	47·27	151·4	987·8	1139·2	8·0
11·63	188·2	2,645	0·02366	42·33	156·8	984·7	1141·5	9·0
9·57	193·3	2,399	0·02608	38·37	161·9	981·8	1143·7	10·0
7·55	197·8	2,193	0·02849	35·09	166·5	979·1	1145·6	11·0
5·51	202·0	2,002	0·03089	32·35	170·8	976·5	1147·3	12
3·47	205·9	1,880	0·03332	30·01	174·8	974·1	1148·9	13
1·43	209·7	1,750	0·03572	28·01	178·6	971·8	1150·4	14
Atmosphere 0	212·0	1,672	0·03738	26·75	180·9	970·5	1151·4	14·7
Pounds per square inch above atmosphere. 1·3	216·3	1,543	0·04046	24·71	185 4	967·7	1153·1	16
2·3	219·4	1,456	0·04282	23·35	188·5	965·8	1154·3	17
3·3	222·4	1,381	0·04514	22·14	191·5	964·1	1155·5	18
4·3	225·1	1,315	0·04752	21·04	194·2	962·4	1156·7	19
5·3	227·8	1,250	0·04985	20·06	196·9	960·8	1157·8	20
10·3	240·0	1,002	0·06142	16·28	209·2	953·1	1162·3	25
15·3	250·3	858	0·07283	13·73	219·8	946·4	1166·2	30
20·3	259·2	743	0·0841	11·89	229·0	940·5	1169·5	35
25·3	267·1	655	0·0954	10·48	237·1	935·1	1172·2	40
30·3	274·3	587	0·1064	9·393	244·4	930·3	1174·7	45
35·3	280·9	532	0·1175	8·513	251·0	925·8	1176·9	50
45·3	292·5	448	0·1393	7·175	263·0	917·8	1180·8	60
55·3	302·7	388	0·1610	6·211	273·6	910·6	1184·2	70
65·3	311·8	345	0·1824	5·481	282·9	904·0	1187·0	80
75·3	320·0	307	0·2039	4·908	291·4	898·2	1189·6	90
85·3	327·6	277	0·2250	4·446	299·2	892·6	1191·8	100
105·3	340·9	234·1	0·2667	3·746	313·0	882·4	1195·4	120
125·3	352·7	202·5	0·3085	3·241	325·4	873·2	1198·6	140
145·3	363·2	178·6	0·3450	2·858	336·3	864·9	1201·2	160
165·3	372·8	159·9	0·3910	2·558	346·2	857·3	1203·5	180
185·3	381·6	144·5	0·4322	2·316	355·5	850·2	1205·6	200
210·3	391·6	129·5	0·4824	2·073	366·0	841·8	1207·8	225
235·3	400·8	117·3	0·5328	1·877	375·6	834·3	1210·0	250
260·3	409·3	107·2	0·5828	1·716	384·6	827·1	1211·7	275
285·3	417·0	98·7	0·6329	1·580	392·9	824·3	1213·0	300
320·3	427·2	87·2	0·7168	1·395	402·1	814·4	1216·5	335
350·3	435·7	80·4	0·7770	1·287	410·7	808·3	1219·0	365
400·3	448·9	71·3	0·8764	1·141	424·0	798·8	1222·8	415
450·3	459·0	64·0	0·9756	1·025	435·0	791·1	1226·1	465
500·3	470·0	58·1	1·0741	0·931	446·0	783·5	1229·5	515

CHAPTER III

NATURE AND EFFECTS OF HEAT

The Steam Engine is a Heat Engine.—The steam engine is not only a machine for the transmission of force, but it is also a *prime mover* which receives its energy from a natural source (viz. the combustion of fuel), and is a heat engine. The steam, derived from the combustion of the fuel, is merely a medium for conveying the heat energy derived from the fuel to the engine, where it is converted into mechanical work. For the proper understanding of the steam engine it is necessary to know something of the nature and properties of heat, and the effects produced on water by the addition of heat.

Nature of Heat.—Only one definite statement can be made of the nature of heat: it is *a form of energy*, which can be conveyed under certain well-defined conditions from one body to another body or substance.

Heat is now generally supposed to consist in the rapid vibratory motion of the particles or molecules of which a substance is composed, and in which the more rapid the molecular motion becomes the greater is the energy in the substance. From this it may be considered that heat is a *condition of matter*, and a condition which can be transferred to another substance. The properties of heat are known to us from the effects produced by the transference or flow of heat from one body to another, and, in accordance with natural laws, a flow of heat can only take place from a hotter to a colder body.

Temperature (T or t).—The temperature is a measure of the condition or *quality* of one part of a body in regard to its being more or less hot than another part of the same body or of another body.

Temperature must not be confounded with heat, which is a *quantity*. Thus two bodies of water may be of the same temperature, but the larger body contains the greater quantity of heat, due to its greater weight. One substance does not directly impart its temperature to another substance; but it can transfer some

of its heat, and in this way raise the temperature of the other substance until a common temperature is obtained, which is always lower than the originally higher temperature.

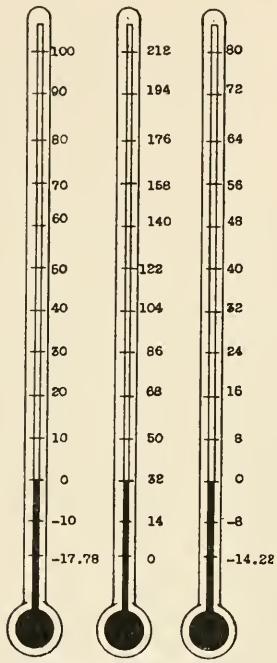



FIG. 10.—Thermometers.

Thermometers.—The instrument used for measuring temperatures is called a thermometer, and the thermometer generally used in engineering, in English-speaking countries, is graduated on the Fahrenheit scale.

The Centigrade, or Celsius, scale is used in countries where the metrical system is adopted, and generally for physical and chemical experiments. The Réaumur system is used in Russia and in some remote parts of Germany; but in both these countries there is a general disposition to adopt the Centigrade system, particularly in Germany.

In each thermometer the boiling and freezing points of pure water, under the normal atmospheric pressure of 14·7 lb. per square inch, are first determined, and the distances between these points respectively for each thermometer are divided up into degrees. The number of degrees correspond to 180, 100, and 80 for Fahrenheit, Centigrade, and Réaumur respectively. The freezing-points are zero for both Centigrade and Réaumur, and 32 degrees above zero for Fahrenheit. Water, therefore, boils (under normal atmospheric pressure) at 212°, 100°, and 80° respectively on the Fahrenheit, Centigrade, and Réaumur thermometers. A diagram of the three systems is shown in Fig. 10.

To convert R. into C. : $R. = \frac{4}{5}C.$, and $C. = \frac{5}{4}R.$
,, C. into F. : $F. = \frac{9}{5}C. + 32$, and $C. = \frac{5}{9}(F. - 32)$
,, F. into R. : $F. = \frac{9}{4}R. + 32$, and $R. = \frac{4}{9}(F. - 32)$

Effects of Heat.—When heat passes from one substance to another, three distinct changes are effected in one, and at the same time similar changes in an opposite sense take place in the other. The heat energy transferred to the substance causes changes in the substance itself. Part is absorbed in (1) doing *internal work*, by increasing the vibratory motion of the molecules and tending to increase its volume. This increased vibratory motion is accompanied by (2) an increase in temperature. Another part (3) is absorbed in doing work on external bodies, generally termed *external work*, but only when an increase in volume is allowed to take place, and thus the resistance or pressure of other surrounding bodies is overcome

through a space equal to the increase in volume. If no change in volume takes place (and this can only happen in the case of comparatively easily compressible substances such as gases and certain liquids, but unusual in metals), and the same quantity of heat passes as before, then a further rise in temperature takes place, and heat is added to the *internal energy* of the substance exactly corresponding to the work done in resisting its expansion.

All substances, under certain conditions of pressure and temperature, can be made to take either the solid, the liquid, or the gaseous state, and therefore, by adding heat to or taking heat away from any substance, either one of the following three effects may be produced: (1) expansion or contraction; (2) liquefaction or solidification; (3) evaporation or liquefaction.

Expansion of Solids and Liquids.—When heat is added to a substance, solid or liquid or gaseous, it expands to a more or less degree corresponding to the difference in temperature. (There are a few exceptions to this rule: india-rubber being, perhaps, the most common in use; water also contracts when raised from freezing temperature to about 39·1° Fahr., but above this temperature expands.) A solid and practically incompressible substance such as a metal expands in three directions—length, breadth, and thickness; but for general purposes of engineering a knowledge of its expansion in one direction is sufficient. It is approximately the same amount per unit of dimension in each of the three directions. The *linear* expansion of solids, or increase in length, is that generally considered and measured.

The *coefficient of linear expansion* is the increase of length per unit of length when the temperature is increased 1°. This coefficient increases with the temperature, but for ordinary purposes may be considered constant in value.

EXAMPLE.—A steel rod, whose length is 1 at 32° Fahr., increases to a length of 1·0011447 at 212° Fahr.: what is the coefficient of expansion?

$$\text{Coefficient} = \frac{0{\cdot}0011447}{180} = 0{\cdot}000006415.$$

Also, what is the increase in length of a steel tube, 7 feet long, when the temperature is increased from 32° to 732° Fahr.?

For each degree, 1 foot becomes $(1 + 0{\cdot}000006415)$ feet
„ 700 degrees, 1 „ „ $(1 + 700 \times 0{\cdot}000006415)$
„ 700 „ 7 feet become $(1 + 700 \times 0{\cdot}000006415) \times 7$
Linear expansion $= 4900 \times 0{\cdot}000006415$ feet
$= 0{\cdot}0314335$ feet
$= 0{\cdot}377$ inch

Liquids expand to a moderate extent, and it is only necessary to measure the increase in volume, which is three times (very approximately) the linear expansion. Thus if x be the coefficient of linear expansion, then the volume of a cube whose side is originally 1 becomes $(1+x)^3 = 1+3x+3x^2+x^3$. The value of $3x^2+x^3$ is so small that it may conveniently be neglected, so that the new volume $=1+3x$ very approximately.

In a mercury barometer, the glass container or tube has a very small coefficient of expansion, while the mercury has a very large one ; the expansion of the glass can generally be neglected, while the mercury, increasing in *volume*, apparently rises three times the amount due to its linear expansion. The above is a very common case with glass vessels and liquids ; but if a metal vessel is used, an actual depression may be sometimes observed in the height of certain liquids when heated, because the coefficient of expansion of the metal vessel is so much greater than (more than three times) that of the liquid.

The difference in the expansion of two metals can be readily observed by riveting the opposite ends of two different metals together, and then heating them ; the difference of expansion produces unequal increases in length between the rivets, and causes some curvature, in which the metal with the greater coefficient takes the outside, and therefore longer, curve.

Allowance for expansion (actually for unequal expansion of two bodies) is very common and almost universal in engineering structures. The rails for tram and other railways are generally placed with their ends a certain distance apart, so that when warmed by the sun they can expand lengthways ; similarly, steam pipes are fitted with expansion glands or with bends to allow expansion and contraction. Iron bridges, boiler furnaces, and some boiler stays have also some arrangement by which expansion is allowed within certain limits. Smoke-box doors and boiler casings, when made flat, tend to warp when heated, and are now frequently constructed of corrugated plating, so as to maintain their original form and shape.

In the foundry, the cooling of molten metal when run into a mould frequently causes fracture from unequal contraction ; for this reason the spokes of a large cast-iron or steel wheel are curved, so that contraction is allowed for when the metal cools. The pattern used for moulding must also be larger than the article to be made. Boiler furnaces and tubes, cylinders and their connections, are subjected to change of form when heated and cooled, and neglect in allowing for the consequent expansion or contraction has frequently resulted in fracture.

Water expands when cooled from 39·1° Fahr. downwards until ice is formed,

when further expansion takes place. Ice forming in the mouth of a pipe prevents further expansion of the water within it, and the pipe may burst. For this reason water pipes are generally laid at some depth below the surface of the ground, where they are less subject to cooling effects or changes of temperature and state.

Expansion of Gases.—Unlike solids and liquids, gases expand easily and infinitely, and thus any space or volume to which a gas has access is immediately filled with gaseous vapour; or, when a portion of the gas contained in a chamber is removed by the action of a pump or other means, the gas remaining in the chamber will instantly expand so as to fill the whole volume. Conversely, gas or air can be compressed so as to occupy comparatively very much smaller volume.

In considering the volumetric expansion of air or gas within an engine cylinder, the expression *stroke volume* is commonly used or implied. Referring to a cylinder and piston, such as are shown in Fig. 11, it will be seen that the volume of the gas inside the cylinder and behind the piston = length $AC \times$ area of piston. If the piston be moved to any point, G, the volume = length $AG \times$ area of piston. Thus the volumes of the gas at C and G respectively are represented (for the same cylinder) by the lengths AC and AG. In other words, if the stroke of the piston be represented by any line, AB, it also represents to scale the volume, depending on the area of the piston, which is constant for the same cylinder. Also, if any two points, C and G, be taken in this line, the distance between them represents, to scale, both the stroke of the piston and the volume struck out by it between the two points.

The **First Law of the Permanent Gases,** commonly known as Boyle's or Marriotte's law, states that—

"The volume (V) of a given *mass* of a gas varies inversely as the (absolute) pressure (P) provided the temperature (T_0) be kept constant."

Briefly, this law means that if 2 gallons of a gas are compressed into 1 gallon, provided no change in temperature takes place, then the absolute pressure will be doubled; and so on. Using the usual notation—

$$V \propto \frac{1}{P} \text{ when } T_0 \text{ is constant,}$$

$$P \,.\, V = \text{constant when } T_0 \text{ is constant,}$$

and

$$P \propto \frac{1}{V} \text{ when } T_0 \text{ is constant.}$$

Boyle's law of expansion can be represented graphically. Let us suppose that a given mass of gas fills a certain space behind a piston in a cylinder, as shown in Fig. 11, and take a straight horizontal line, AB, to represent the extreme stroke of the piston to some convenient scale, and suppose the piston in the first instance to be at C, so that AC represents the volume of the given mass of gas. From A set up AD perpendicular to AB, and representing the absolute pressure of the gas in the first instance, termed the *initial absolute pressure*, to some convenient scale; so that 1 inch in height of AD represents a certain known pressure per square inch, and so on for each inch and for each part of an inch. Describe a rectangle, $ADEC$; then the area of this rectangle is equal to—

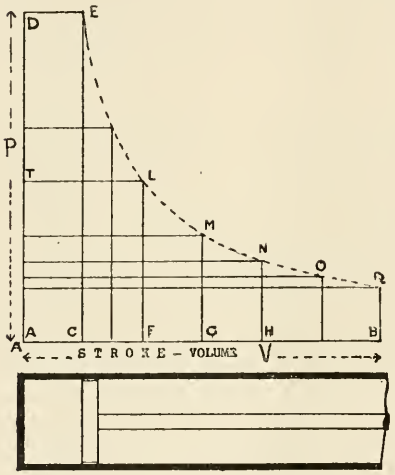


Fig. 11.—Boyle's Law and Stroke-volume.

$$AD \times AC = p \,.\, v.$$

Cut off CF, and make it equal to AC, so that when the piston is moved to F the gas inside the cylinder expands to twice its original volume. Make $AT = \frac{1}{2}AD$. Then the rectangle $ATLF$ = rectangle $ADEC$ in area, and the line FL represents the absolute pressure when the piston is at F. Any number of rectangles may be described as shown in the figure, each of which is equal to $ADEC$, and a series of points, E, L, M, N, O, Q . . . may be found such that their respective vertical heights above AB represent to scale the absolute pressure at those points for the respective volumes AC, AF, AG, AH. . . . By drawing a fair curve through the points E, L, M, N, O, Q . . . a graphical representation is obtained of the absolute pressure in the cylinder for every point in the stroke of the piston.

Thus, $p \,.\, v = AD \times AC = BQ \times AB = P \,.\, V = \text{constant}.$

It must not be forgotten that this law assumes that the expansion is *isothermal* (*i.e.* that no change of temperature takes place as the gas expands). As we shall see later, no work is then done by the expanding gas when the piston moves, unless heat is received into the gas from some outside source.

Gases also expand when heated, or tend to expand, and if the tendency to

expand is resisted by the containing vessel or other means, then the pressure is increased because the volume cannot increase. Conversely, if heat is taken away from a gas the pressure decreases, because the gas expands and keeps the volume constant by filling the same space.

Absolute Temperature.—For every decrease of 1° in temperature of air below the freezing-point (32° Fahr.), it loses about $\frac{1}{493}$ part of its volume, and therefore it is assumed that at a temperature of 493° below 32° Fahr., or of 461° below the *zero* of the Fahrenheit thermometer scale, both volume and temperature disappear. This point is taken as the *zero of absolute temperature.* For example, the absolute temperature of the boiling-point of fresh water under ordinary atmospheric pressure is—

$$212° + 461° = 673°.$$

We are now in a position to state the **Second Law of the Permanent Gases,** as formulated by Charles and by Gay-Lussac :—

"Under constant pressure, equal volumes of different gases increase equally for the same increment of temperature. Also, if a gas be heated under constant pressure, equal increments of its volume correspond very nearly to equal intervals of temperature as determined by the scale of a mercury thermometer."

Or $V \propto T_0$ when P is constant, and T_0 is the *absolute* temperature.

Thus if 1 gallon of gas is heated until its absolute temperature is doubled, its volume is also doubled, becoming 2 gallons if no change in pressure takes place.

For a given mass of gas, by combining the first and second laws,

$$P \,.\, V = c \,.\, T_0,$$

where c is a constant depending on the specific density of the gas, and on the units in which P and V are measured.

If P is measured in pounds per *square foot*, V is the volume of 1 lb. (weight) of the gas in cubic feet; and T_0 is the absolute temperature on the Fahrenheit scale.

Then, for air—

$$P \,.\, V = 53{\cdot}18 \; T_0 \text{ in foot-lb.}$$

EXAMPLE.—At 32° F., P = 14·7 lb. per square inch = 2116·8 lb. per square foot.
V = 12·38 cubic feet, by direct experiment.

Therefore $P \,.\, V = 26{,}217$ foot-lb.

From which, when T_0 = 32° Fahr. = 493° absolute temperature Fahr.—
$c = 53{\cdot}18$.

It follows from the above that when air, or gas, or steam-gas is

heated in a closed cylinder, so that its volume cannot change, an increase in pressure takes place, and the internal (or intrinsic) energy of the gas is increased.

Also that when air, or gas, or steam-gas is heated in a cylinder closed by a movable piston on which a constant pressure is exerted, an increase in volume takes place, and, in addition to the increase in internal (or intrinsic) energy of the gas, external (or visible) work is done by moving the piston against a resistance through a certain distance obtained by the increase in volume.

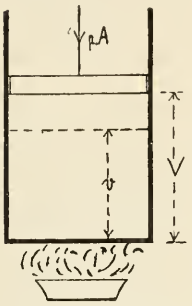


Fig. 12. — External Work during Expansion.

Let v = the original volume,
p = constant pressure per unit of area A,
V = the final volume.

Then (see Fig. 12)—

External work done
= p . A × distance piston is moved
= p(V − v)
= pressure × change in volume.

As previously stated, for each increase in temperature of 1° Fahr., air expands $\frac{1}{493}$ part of its volume ; and this ratio is found to be approximately true (although not exactly) whether the increase be from 299° to 300°, or from 60° to 61° Fahr., or at any temperature in ordinary use. The coefficient of expansion per unit of volume of ordinary dry air is therefore = $\frac{1}{493}$ = 0·002 approximately.

Sensible Heat (S).—When a change of temperature alone takes place in a substance, the quantity of heat used in making the change is called *sensible* heat.

For water, the sensible heat is the quantity required to raise the temperature of 1 lb. weight of water from 32° F., or other temperature, to the point of ebullition, or *boiling-point* ; a change of state then begins to take place, and no further increase in temperature occurs (unless the boiling-point is raised by increasing the pressure) until the change of state is completed. Thus a substance may continue to receive heat at certain temperatures, peculiar to the substance and the pressure to which it is subjected, without changing its temperature.

Quantity of Heat.—Various substances have varying capacities for taking in or giving out heat ; it requires more or less fuel to raise the temperature of one substance 1° than another. Thus it will cost more fuel to raise the temperature of 1 lb. weight of water through 1° than it will to raise the temperature of the same weight of steel

through 1°. It is necessary to have some standard of comparison, and this is the **British Thermal Unit** (B.Th.U.), which is the quantity of heat required to raise 1 lb. weight of pure water at its maximum density (39·1° Fahr.) 1° in temperature. For all practical purposes this quantity may be considered as constant, whether the water be raised from 39° to 40° or from 300° to 301°.

The quantity of heat required to raise unit mass of a substance 1° in temperature is called the **Specific Heat** of the substance.

As we have seen above, the specific heat of water is 1, or unity, and 1 b.t. unit is required to raise 1 lb. of water 1° in temperature. Therefore, in engineering, the specific heat of a substance is the quantity of heat in b.t. units required to raise 1 lb. of the substance through 1° in temperature.

A simple method of determining the specific heat of a substance is to immerse a known weight of it at a known temperature in a known weight of water at a differing temperature. After the two substances have reached the same temperature by communicating heat one to the other, the heat lost in one must be exactly balanced (assuming no external loss takes place) by that gained in the other, and an equation is obtained—

$$\text{Weight} \times \left\{\begin{matrix}\text{rise or fall in}\\ \text{temperature}\end{matrix}\right\} \times \left\{\begin{matrix}\text{spec. heat}\\ \text{of subst.}\end{matrix}\right\} = \text{weight} \times \left\{\begin{matrix}\text{fall or rise in}\\ \text{temperature}\end{matrix}\right\} \times \left\{\begin{matrix}\text{spec. heat of}\\ \text{water (1).}\end{matrix}\right.$$

For accurate measurement and exact determination of the specific heat, various corrections must be made to compensate losses, such as heating or cooling the containing vessel and instruments used (thermometers, etc.).

Changes of State.—The quantity of heat required to change 1 lb. of a substance from a given state into another state without altering its temperature, is called the *Latent Heat* (L) of the substance. For each substance there are two latent heats, corresponding to the two changes of state respectively from the solid to the liquid (or latent heat of fusion) and from the liquid to the gaseous (or latent heat of evaporation); and conversely.

Suppose it is required to change 1 lb. of ice (specific heat of ice = 0·5 approximately) at temperature 0° Fahr. into steam at 212° Fahr., under normal atmospheric pressure; then—

			B.t. units.
Change of temperature,	sensible heat, at 0·5 per 1°,	from 0° to 32° F. =	16
Change of state,	latent heat (of ice),	at 32° F. =	144
Change of temperature,	sensible heat,	from 32° to 212° F. =	180
Change of state,	latent heat (of water),	at 212° F. =	966
		Total quantity of heat required =	1306

It should be noticed that a change of state and a change of temperature do not take place at the same time under the generally accepted engineering conditions, but some changes of state take place at all temperatures in nature; these

changes, are, however, extremely slow, and do not enter into the present discussion.

Regnault gives 142 b.t. units as the latent heat of fusion of ice. Later experiments by Bunsen and others give the higher value of 144, which is that now generally used by refrigerating engineers. The American Society of Mechanical Engineers recommends a standard unit of ice-melting capacity, equivalent to 288,000 b.t. units per ton (equal to 2000 lb. British weight).

The value 966 is the approximate result of experiments by Regnault.

Evaporation, Ebullition, Boiling-Point. — *Evaporation* is said to take place when vapour is slowly formed at the surface of a liquid, and is discharged into a space not already saturated with such vapour. The rate of evaporation is principally influenced by (*a*) the difference in temperature of the liquid and its surroundings; (*b*) the quantity or degree of saturation or density of similar vapour in the atmosphere or other substance in contact with the surface of the liquid; (*c*) the superincumbent pressure in the space due to other vapours; and (*d*) the area of the surface at which evaporation can take place.

Ebullition, or boiling, takes place when elastic bubbles of vapour are formed in the liquid itself, and are discharged into a space already saturated with similar vapour. The kettle "sings" when the heating releases the air, or other gases, previously in solution in the water. The escape of these bubbles assists materially in the formation of steam, and, heat continuing to be added, ebullition, or boiling, begins shortly after the singing begins.

The temperature of ebullition, or the *boiling-point*, increases with the pressure,[1] and for a given pressure in any one liquid the boiling-point is always the same. The boiling-point remains unchanged, whether the heat is applied either quickly or slowly or with great intensity. The boiling-point of water under different pressures has been determined by several series of experiments, principally by Regnault, and these are shown in column 2 of the table at the end of Chap. II.

Properties of Vapours.—Atmospheric air particularly and gases generally have the property of absorbing vapours given off from

[1] When the surface of the water is open to the ordinary atmospheric pressure of 14·7 lb. per square inch, fresh water boils at 212° Fahr.; but if the vessel be entirely closed, so that the steam pressure increases, the boiling-point rises, and at about 21 atmospheres' pressure (=300 lb. per square inch), such as now used in Naval boilers, the boiling-point is about 422° Fahr. The atmospheric pressure decreases generally with increased height above sea-level, and water then boils at a lower temperature; thus on top of Mont Blanc (height 15,000 feet) it is at about 183° Fahr. The "boiling-point" also increases with the density of the liquid: thus ordinary sea water, which is of greater density than fresh water, boils at 213·2 Fahr.—not 212° Fahr.—under normal atmospheric pressure.

liquids at temperatures below the respective boiling-points of such liquids. The air or gas into which a vapour escapes from the surface of a liquid is not capable of holding up more than a certain quantity, and when it has taken up this quantity the air or gas is said to be saturated with the vapour of the liquid. Thus the atmosphere, when only partially saturated with water vapour, will continue to drink up water vapour at any ordinary temperature either at or below that of the boiling-point. This process is continually going on in nature, and moisture is absorbed from everything which yields it —earth, plants, and animals. Generally, the proportion of vapour required to produce saturation increases as the temperature of the absorbing gas or air increases; and the rate of absorption increases with the temperature at which the vapour is formed. Any decrease in temperature of the absorbing gas or air, when saturated, produces a condensation of vapour which falls in beads of moisture to the earth or other substance beneath.

Evaporation, or formation of vapour, below the boiling-point is very common with alcohol, spirituous oils, and petroleum oils; the particles thus escaping from the surface of the liquid exist in a free state as vapour. Dry air will thus take up, and retain until saturated, the vapours from petrol motor spirits of about double the quantity at a temperature of 80° Fahr. than at 50° Fahr.; and with certain pressures and proportions of volume, in a closed space the mixtures thus produced become explosive.

Above a certain *critical temperature*, which is peculiar to each vapour and corresponding generally to its chemical composition, the *vapour* becomes a *gas*, and cannot be condensed by *compression alone*. At the critical point the specific volumes of a liquid and of its saturated vapour are equal. This critical temperature for steam and water vapour is above that usually obtained in steam-engine practice, and need not be considered here; but bearing it in mind, it may be stated that in practice any compression of the saturated vapour of water (saturated steam) will thus produce a condensation of some of the particles of which it is composed. Water, or any liquid, contained in a closed vessel and only partially filling it, will give off sufficient vapour to fill the remaining space within the vessel from which the air has been wholly or partially withdrawn.

Vapours are also given off by some solid bodies without apparently passing through the intermediate liquid state. Thus ice gives off vapour, which is absorbed by the atmosphere; also coal, especially when of a gaseous or bituminous nature, gives off hydrocarbon gases which, under certain conditions of temperature and proportionate mixture with the oxygen of the air, become spontaneously ignited and sometimes explosive. (See also "Spontaneous Ignition," Chap. X.)

Transference of Heat.—Heat is transferred from one body to another in three ways—by conduction, by convection, and by radiation; either together, severally, or separately. In any case, heat can only be transferred from a hotter to a colder body.

In a boiler, for example, as shown in Fig. 13, the fire radiates heat to the furnace and some parts of the combustion chamber; heat then passes through the plating by conduction; and the water is heated and steam formed by convection. The natural draught up the chimney or funnel is caused by convection, and thus draws in the air necessary for combustion from the ashpit.

Conduction occurs when heat passes from one part of a body to another part of the same body, when it may be termed a case of *internal* conduction; or it may occur when heat passes from one body to another in contact with the first, when it may be termed a case of *external* conduction. The rate of conduction is always greater in internal than in external conduction, principally because it is impossible to obtain perfect contact between two bodies. The nearest approach to perfect contact is probably in alloys of metals, when the rate of conduction is always less than that of the internal rate of the better-conducting metal separately.

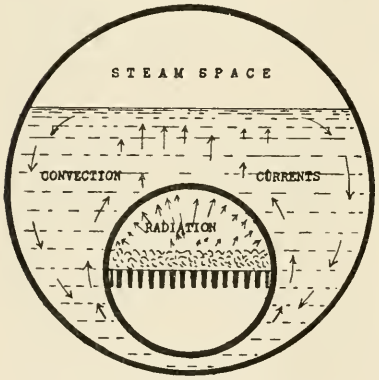


FIG. 13.—Transference of Heat.

Many substances conduct heat very slowly, and are then called *non-conductors* of heat; gases, fluids, earthy substances (such as brick, porcelain, marble, and porous substances generally), are bad conductors, or relatively non-conductors of heat. Steam pipes, cylinders, exteriors of boilers, etc., which should be kept warm, are usually covered with some non-conducting material, commonly called *lagging*, so that as little heat as possible shall escape. For the same reason, bodies which should be kept cool, such as ice, are also covered to prevent heat passing into them from outside influences. In all steam or heat plants the lagging is of great importance, and great attention is paid to its proper provision and repair, as, quite apart from possible inconvenience to persons near, the efficiency of the lagging has an important bearing on the economical results which may be obtained from the engine and plant generally.

Convection.—In fluids and gases heat is transferred internally by convection currents, which practically means a circulation of the heated particles upwards, some heat being transferred to the particles in the first instance by either conduction or radiation. As nearly all liquids and gases expand when heated, the heated particles become lighter than the colder ones, per unit of volume, and their specific gravity becomes less; these lighter particles rise, and their place

is taken by heavier and colder ones until the general mass of the fluid becomes heated by the continuous circulation of the convection currents. The application of heat to the upper surface of a fluid merely raises the temperature of the fluid at the surface, and practically no convection takes place ; and thus no change of temperature takes place in the lower strata of the fluid. Therefore when a gas or liquid is to be heated, the heat is applied as low down as possible relatively to the fluid, and consequently the furnaces of boilers are generally placed as near the bottom of the boiler as practicable.

For *heating* rooms and buildings, an efficient form of convection heater, shown in Fig. 14, is sometimes used, and consists of a pipe, the

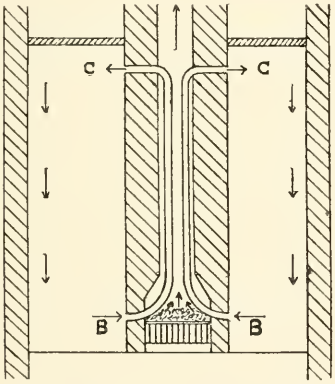


FIG. 14.—Convection Heating (Air).

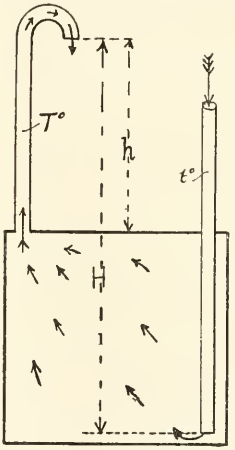


FIG. 15.—Convection (natural) Ventilation.

body of which is in the chimney space, with its opposite ends opening into the room near the floor at B, and near the ceiling at C. Heat from the ordinary fire in the grate warms the pipe (within the chimney), which heats the air inside the pipe ; the hot air rises and escapes into the upper part of the room, while colder air enters the pipe at the bottom to take its place. When properly arranged on this system, large buildings or rooms can be quickly warmed with a very small fire, or source of heat, without vitiating the air, and with less draughtiness than under the usual conditions.

Ventilation by natural means is always obtained by convection, and to obtain any practical efficiency it is necessary to provide a means of inlet and outlet for the convection currents. The ordinary

watertight or other compartments into which a ship is divided can be most efficiently ventilated by natural means, *without the cost of energy or power*, by providing an outlet, or uptake, in the roof of the compartment, and an inlet, or downtake, to bring the colder air down to almost the bottom of the compartment, as shown in Fig. 15. If the temperature of the air in the compartment is greater than the external air, the upcast current, set in motion by convection, continuously draws in a cold current through the downtake, and the greater the difference in temperature, the more rapid (increasing with the square root of the difference in temperature) becomes the flow of cold air into the compartment. As the hotter air occupies more space than the colder, the area of the uptake opening should generally be greater than that of the downtake—usually $1\frac{1}{4}$ to $2\frac{1}{2}$ times.

The velocity, or rate of flow, of the air upwards through a ventilator, or a chimney, is dependent on the head of pressure created by the difference in weight of two columns of ascending and descending air *of the same height as the ascending column*. The only influence which the descending column has on the rate of flow is by virtue of its lower temperature, and to obtain a low temperature it is necessary to tap the supply from a cool place. *Cowls* fitted above supply ventilators may increase the rate of flow by concentrating the air and increasing its density or pressure, thus making it heavier, and this density or pressure is greatest when the temperature is the lowest available. When ventilation is most required—in tropical harbours, for example—cowls are sometimes inoperative, because by introducing the cold air near the top of a compartment, the virtual height of the ascending column is decreased and the air remains stagnant in the lower parts of the compartment.

The velocity (V) in feet per second, due to certain conditions of height (H) of the ascending column, and difference in absolute temperature of the ascending (T_0) and descending column (t_0), is—

$$V = \sqrt{2g \cdot \frac{T_0 - t_0}{t_0} \cdot H},$$

where g = acceleration due to gravity = 32 approximately.

If the absolute temperature of the descending air is about 500° (= 39° Fahr.)—

$$\frac{1}{t_0} = \frac{1}{500} = 0{\cdot}002,$$

which is a value near enough for all ordinary cases of ventilation.

To take an example, if we allow for a rise of temperature of 10° Fahr.—

$$V = \sqrt{64 \times 0{\cdot}002 \times 10 \times H} = 1{\cdot}1\sqrt{H} = \sqrt{H} \text{ nearly}$$

if some allowance is made for frictional losses in the ventilators.

Next, suppose the height of a compartment is 21 feet, $(H - h)$, and the height of the uptake ventilator, h, is 16 feet above this; then the rate of flow is—

$= 4$ feet per second, when the supply comes in at the top,
or $= 6$,, ,, ,, bottom,

and the rate of flow is considerably increased by any further rise in temperature of the compartment.

A man requires about $\frac{1}{2^{\cdot}0}$ cubic foot of free air per second, and by a further simple calculation it can be shown that a sufficient supply could be obtained under the above conditions, and neglecting frictional resistances, through a hole only about $\frac{1}{2}$ inch in diameter, the supply coming in near the bottom and the uptake being 16 feet high above the compartment. This hole seems very small, but when a diver's supply pipe is examined it is seen that it is not far from the truth, although the means adopted to ensure the supply differs in method but not in principle; each is caused by a difference in density, or head of pressure—the diver's supply by artificial means, and the compartment by natural means.

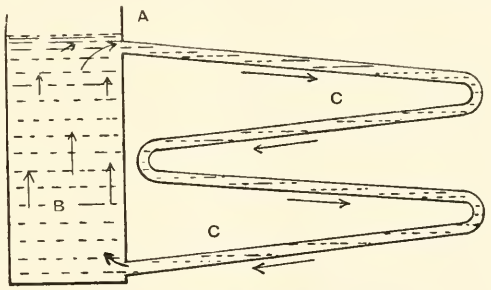


FIG. 16.—Convection Heating (Water).

The practical difficulties of ventilation generally arise from a misconception of the simple principle involved, and instances are very common where the uptake is practically non-existent (except either in the form of an uncovered hatch, or of a very short bent pipe), and where the supply also enters at the same level; very little natural ventilation can occur under these conditions.

The convection currents which are set up in water when heated are utilised for *heating rooms*, houses, and ships, and in all cases the principle is precisely similar to that in a boiler, although there is necessarily some difference in constructive detail. A simple arrangement is shown in Fig. 16, and is frequently used in the common or garden greenhouse. There is a vertical boiler, B, with an open top or a removable cover, and the fire is lighted either immediately beneath the boiler or in a furnace fitted in its lower part. The convection circulates the

water upwards and then away at A, through the pipe C shown, the opposite end of which is brought back to the boiler at the bottom. The hotter water communicates its heat by conduction from the interior to the exterior of the pipe ; heat is radiated from the outer surface to the objects in the greenhouse, and the water, thus becoming cooled, returns downwards to the boiler.

For heating other places, it is generally necessary to entirely close the boiler so that it is pressure- and water-tight, and to carry the piping vertically upwards to the highest point necessary for the system and place. The boiler and run of pipes must be kept entirely filled with water, and this is done by fitting a cistern or supply tank, with an automatic ball-valve arrangement, above the highest point to which the water can be circulated. An overflow pipe is also necessary, and is fitted also near the highest point, to prevent overloading or oversupply. A common example of such a system is found in most houses for supplying hot water from a tank fitted in the top of the house, and connected by two pipes, upcast and downcast, with a boiler fitted in rear of the kitchen fire.

In all systems of hot-water heating (as in water-tube boilers also) certain precautions are necessary. The interior of the boiler should be accessible for periodical cleaning ; the pipes should incline upwards for the upcast current and downward for the downcast current (vertical, if possible) ; and near the boiler, where scale is likely to accumulate, the pipes should be vertical and arranged so that the scale, when loosened, should fall into the boiler and not be lodged in the pipes and so choke them. At no part of the pipe system should any part be horizontal, and sufficient inclination should be given so that when the boiler is emptied the whole system is completely drained. If, as on board ship, such complete drainage is difficult (although never impossible), owing to the lack of head room, then drain cocks should be fitted to each pocket so that complete drainage can be obtained in this way. It is evident that circulation by convection in a horizontal pipe cannot occur from this cause alone, and is only obtained from some difference in temperature and level at other parts of the system. If properly designed, even in a ship, self-drainage, without pockets and independent drainage, is quite possible of attainment, and should be the rule and not the exception.

Referring to Fig. 16 it should be noticed that if the fire be applied beneath the tubes, instead of beneath the boiler, the circulation by convection is reversed, and the system becomes virtually a water-tube boiler, in which the original boiler becomes a downtake pipe, as shown in Fig. 17.

A good natural circulation, which can only be obtained by convection, is essential to the safe working of all boilers. In recent years, water-tube boilers have been gradually developed into two distinct classes : (1) those in which the tubes are nearly horizontally arranged, as shown in Fig. 17 ; and (2) those in which the tubes are nearly upright as shown in Fig. 18, and in which the circulation is naturally rapid. With a slow circulation, the heat conveyed to the tubes is taken away slowly by the sluggish movement of the water ; bubbles of steam, which is practically a non-conductor of heat, are formed, and, not escaping quickly enough, overheating of the material of the tubes is therefore probable when more than a moderate quantity of heat is given out by the fuel from the furnace. A rapid circulation of water, as in a boiler with nearly upright tubes,

allows a maximum quantity of fuel to be burnt, and a maximum quantity of steam to be formed, without any tendency to overheat the material; but, as in all boilers, only for so long as the water side of the heating surface remains clean and free from grease and other deposits of scale.

The figures are typical of the direction of circulation by convection, and do not exactly represent any particular boilers. Every boiler is a study in itself, and a conclusion should not be drawn either that the boiler shown in Fig. 17, for example, is necessarily unsafe, or that that shown in Fig. 18 is comparatively a great improvement, without a full consideration of other merits or disadvantages, in conjunction with the proper allowance for unequal expansion of the material, the weight, the space, and the work which the boiler is expected to do.

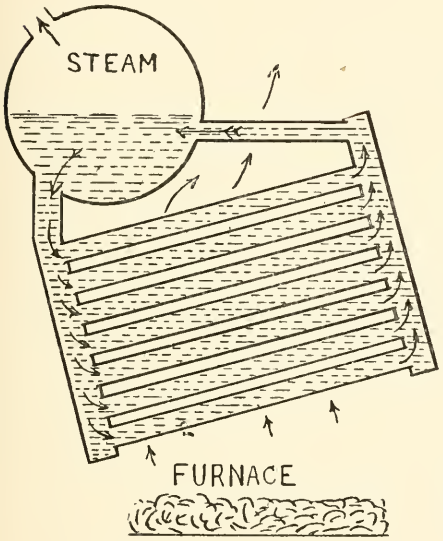


FIG. 17.—Circulation in Boiler.

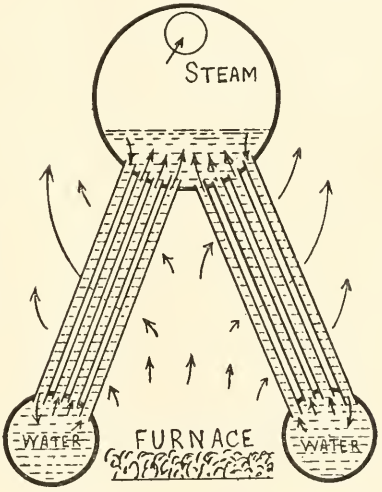


FIG. 18.—Circulation in Boiler.

The temperate climate of the British Isles, and also that of Japan in a lesser degree, is due to convection in conjunction with certain *ocean currents*, the upper surface of which, being always warm, maintains an equable temperature in the shallower water which surrounds the coasts. These ocean currents are caused, in the first instance, by the rotation of the earth as a globe about its axis, pitched nearly at the poles, north and south. The solid part of the world, the land, rotates with uniform angular velocity about this axis, from west to east; but the ocean, being fluid and not solid, does not keep pace exactly with it. Near the equator, where the earth has its greatest velocity (not angular velocity, which is equal from pole to pole), the water fails

to keep pace in the greatest degree, and causes a steady and apparent current or flow from east to west, generally called the equatorial current. Near the poles, where the earth has little or no velocity, there is practically no current caused in this way. The direction which the equatorial current takes is generally determined by the contour of the coast-lines, but is also assisted by the varying depth and formation of the bed of the ocean. The surface water of the ocean near the equator is always warmed by the radiant heat of the sun, and this warmer water, tending to always remain near the surface, accounts for these currents continuing to be warm long after they have left the equator and have almost reached the poles.

The Gulf Stream is a notable instance of an ocean current, warm water being carried from the Gulf of Mexico, past the Florida peninsula, northward and eastward towards Iceland and the North Pole, then southward round the British Isles, and is finally lost in the equatorial current. The colder water flows in varying direction beneath the warmer current, and thus tends to maintain a constant level. The colder current sometimes reaches the surface, a particular case being evident in a southward current moving along a narrow belt off the east coast of North America until it apparently plunges below the warmer Gulf Stream.

Heat is imparted from one body to another and colder one *through space* by *radiation*, in exactly the same way that light is radiated from a luminous body. The rays of heat are sent out in every direction in straight lines, invisible to the eye, and the intensity varies very nearly *inversely* as the square of the distance between the two bodies, one giving out and the other receiving heat. It is in this way that a common fire imparts its heat to the walls and other objects in a room, and also in the same way that a boiler fire communicates some of its heat to the furnace plating and sides. A rough body radiates more heat, or receives more radiant heat, than a smooth or polished body (principally because the surface area is greater in rough than in smooth bodies), and therefore a polished kettle or a steam pipe will radiate less heat than one which is rough and unpolished; but the oiling of a smooth surface increases its power of radiation.

The rays of heat received by the earth from the sun are absorbed by the intervening atmosphere to some slight extent, and by moisture in the atmosphere to a greater extent. The intensity of the heat radiated to the earth is diminished by clouds of vapour in the atmosphere and by the relative density of the atmosphere. Vapour rising from the earth is gradually reduced in temperature with the reduced pressure of the atmosphere at greater height (its boiling-point is lowered, but its capacity to hold vapour is also decreased), and, condensation eventually ensuing, beads of moisture are formed. This form of moisture,

being of greater weight or density than the mixture of air and vapour, is either re-evaporated at a less height, or deposited as dew, rain, hail, or snow on the earth. A continual cooling of the upper strata of the atmosphere is therefore always taking place, and at great heights, such as the tops of mountains, moisture is deposited at a low temperature, producing, in well-defined instances, perpetual snow.

Comparative Heat Properties of some Substances

	Specific Heat.	Radiating or absorbing.	Reflecting.	Conductivity (k) per cent.		Linear Expansion. Coefficient for 1.
				At 32° F.	At 212°.	
Aluminium	0·202	...	...	34·3	36·2	...
Antimony	0·051	...	...	44·2	39·6	0·00000617
Brass	0·094	11 (dead polish)	89	20·4	25·4	0·0000106
Brass (condenser tubes)	...	7 (bright)	93	...	...	0·0000104
Carbon	0·238 at 60° F.	...	...	...	...	...
Carbon	0·456 at 1750° F.	...	...	...	...	...
Copper	0·092	5	95	71·9	72·2	0·0000101
Iron, cast	0·13	25	75	16·6	16·3	0·0000065
Iron, wrought	0·11 to 0·13	23	77	...	...	0·00000681
Steel	0·116	18	...	...	...	0·00000668
Lead	0·032	...	...	8·5	...	...
Lampblack (soot)	...	100	...	8·36	7·64	0·0000159
Mercury	0·032	...	...	1·48	1·89 at 110° F.	...
Nickel	0·108	...	...	14·0	...	...
Platinum	0·32	...	...	8·4	...	0·00000484
Silver	0·56	3	97	96	...	0·0000112
Sulphur	0·184	...	...	...	...	...
Tin	0·56	...	...	15·28	14·23	0·0000132
Zinc	0·094	...	...	30·3	...	0·0000173
Ice at 32° F.	0·504	...	...	...	...	0·0000286 from −17° to 30° F.
Water at 39·1° F.	1·00000	100	...	0·136%	...	...
Water at 212° F.	1·0130	...	...	...	...	...
Water at 417° F.	1·0500	...	...	...	...	...

Mechanical Equivalent of Heat. — The following laws (of thermodynamics) are sufficiently true for all practical purposes, and for very close investigation, and are the results of many careful and exhaustive experiments by Rumford, Joule, and others :—

First Law.—When mechanical energy is produced from heat, a definite quantity of heat goes out of existence for every unit of work done ; and conversely, when heat is produced by the expenditure of mechanical energy, the same definite quantity of heat comes into existence for every unit of work spent. This may be written—

Heat supplied = gain in internal energy + work done.

In other words, the energy expended in raising the temperature of 1 lb. by weight of water 1° (Fahr.) is capable of doing 780 foot-lb. of mechanical work; and conversely, by expending 780 foot-lb. of work, the temperature of 1 lb. by weight of water can be raised 1°: one British Thermal Unit being then produced. We then say that the "mechanical equivalent of heat" is 780 foot-lb., or—

1 British Thermal Unit = 780 foot-lb.[1];

and is commonly known as Joule's Equivalent = J = 780.

Second Law.—It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.

In accordance with the first law, it would be possible for the whole of the heat energy in a substance to be converted into mechanical energy, but the second law sets a limit on the quantity which can be so converted, and a certain quantity is left unconverted and is necessarily "rejected" as heat. Consequently, the efficiency of the conversion, or of the machine effecting the conversion, must be less than unity, and $= \frac{\text{heat converted into work}}{\text{heat taken in by the engine}}$. In Chap. II. *work* and *energy* were defined, and the unit of measurement was stated as 1 foot-lb. If expressed in heat units, this unit of 1 foot-lb. becomes $\frac{1}{780}$ of 1 British Thermal Unit. It should be clearly understood that no work can be done by an engine without a corresponding fall in pressure of the air, gas, or liquid used to produce motion. This fall in pressure must always be accompanied by a corresponding fall in *temperature*. In all cases the internal energy, or stock of energy, contained in the substance is decreased by exactly an equal definite quantity as the work done.

Energy cannot be dissipated or lost. If all the available energy contained in a substance apparently disappears, and only some part of it is converted into work, then the remainder has been transformed into some other form of energy, and still exists in this other form. Commonly, the energy thus apparently wasted disappears as heat, which is usually produced by friction and diffused into the atmosphere or other surroundings.

The law of the **Conservation of Energy** states: "That the total energy of any one form to which the energy of a given system

[1] Various experiments by Joule, 1849–78, made this value 772-774
,, ,, Rowland, 1879, ,, 778
,, ,, Schuster, 1894, ,, 775
,, ,, Rowland 1895 ,, 778·3
,, ,, Osborne-Reynolds, 1897, ,, 777
The British Association recommended, in 1896, the adoption of 780.
The number 780 is a very convenient one, being divisible by 2, 3, 4, 5, 6, and 13, and by various multiples of these numbers.

of bodies is reducible, is unalterable so long as the system is not acted on from without." This law is not only true of mechanical energy, but of all kinds of energy, such as those of heat and electricity. In all that follows relating to the conversion or transference of energy from one form to another form, this law is the fundamental basis, and should be thoroughly understood.

For example :—Potential energy is supplied to a ship in the form of coal or other fuel. Heat is generated by combustion of the fuel, and some of it is converted by the engine into mechanical energy which is used in rotating the propeller. There are various losses (briefly summarised under the heading of *Efficiency*) during the various processes of conversion, and, finally, the whole of the potential energy originally contained in the fuel is returned to the earth as heat energy.

The Natural Sources of Energy are many, and in nearly all cases the sun is the original source, energy being derived by its action on various natural substances. Thus the heat from the sun produces—

1. *Wind energy*, derived from the convection currents in the atmosphere, producing varying densities, with difference of pressure or force per unit of area. The colder air moves along near the earth's surface, to take the place of the hotter and ascending air, and produces wind.

2. *Water energy*, derived from evaporation of water, the vapour of which, mingling with the atmosphere, forms clouds, from which rain, etc., is deposited on the earth. Water deposited on heights above the ordinary sea-level possesses a certain potential energy, and when flowing to the lower level this energy can be utilised in driving water turbines or other water-driven motors.

3. *Heat energy*, derived from certain decompositions of matter, resulting in the formation of coal, oil, and other fuels, which, when burnt, develop energy by combustion in the form of heat.

4. *Food energy* (actually a form of heat energy), from the growth of plants and animals.

Other sources of energy are—

5. *Tidal energy*, considered to be principally derived from the moon's attraction ; and

6. *Electrical energy*, which exists in nature, but has very little value at present for practical purposes, and which is probably some variety of heat energy.

Electrical energy in common use is *not* derived from a natural source, but is obtained by utilising other forms of energy to produce it ; it is therefore a

secondary motive power. Similarly, hydraulic engines when worked by pressure produced, not from a natural source, but by utilising some other form of energy, are secondary motors. Electricity and hydraulic pressure are generally used as mediums for transmitting energy from one place to another, and should therefore usually be considered, not as motive powers, but as a means of transmission of energy ; thus taking the place of some purely mechanical transmission.

Combustion is the chemical union of a combustible or fuel, such as coal, coke, petroleum, naphtha, kerosene, benzine, gasolene, alcohol, and petrol or other spirit, with oxygen ; combustion takes place rapidly only at a high temperature. When such fuel is burnt, large volumes of gas are given off, which, if confined to a small space, allow great pressures to also be produced. From combustion the heat energy utilised in all primary heat engines (steam engines, guns, oil and gas engines particularly) is derived.

For the ordinary steam engine, heat is supplied by the combustion of coal, oil, or other fuel in the furnace, and steam is generated in the boiler. The heat energy contained in the steam thus produced is utilised, as far as possible, in doing work through the medium of the engine ; but some part of the energy is not utilised, and is then termed " heat rejected."

For oil and gas engines there is no boiler, and the combustible, oil, spirit, or gas, is injected into the cylinder itself, with a certain proportionate mixture of air, which contains the oxygen necessary for combustion. The mixture is ignited and burnt into gases ; a high temperature is produced, and a correspondingly high pressure, which, acting on the piston, produces motion and does work.

For guns, certain chemical substances, containing sufficient oxygen for more or less complete combustion, are ignited by detonation, and combustion follows with a rapid formation of gases at a high temperature and a correspondingly very high pressure. This pressure sets the projectile in motion and does work in exact proportion to the heat energy expended within the bore of the gun.

Internal ballistics, which is the name generally given to the science of heat applied to guns, is neither more nor less than the science of heat applied to the steam or other heat engine.

Heat engines—in which the combustion takes place within the cylinder—are known as *Internal Combustion Engines.*

In a compressed-air engine the energy used in the original compression of the air is transferred as heat to the air so compressed. This heat is expended by the compressed air when it is used to do

work on some external body, and in doing so the air loses heat energy in exact proportion to the work done or resistance overcome in doing the work.

Temperatures of Melting Points, etc.

	Fahr.	Cent.
Alcohol, pure	−148	−100
Aluminium	1157	625
Antimony	1146	621
Bismuth	516	269
Brass, condenser tubes	1652	900
Brass, gunmetal	1661	905
Bronze, phosphor	1900	1038
Cobalt	2672	1467
Copper	1929	1054
Iron, forge pig	2228	1220
,, grey pig	2050	1121
,, white pig	2075	1135
,, second cast	2264	1240
,, wrought welds at	2372	1300
,, wrought melts at	2912	1600
,, pure	2776	1525
Steel, high carbon	2400	1315
,, low carbon	2876	1580
Lead	626	330
Mercury	−38	−39
Nickel	2588	1420
Paraffin wax	129	54
Platinum	3227	1775
Silver	1750	954
Sulphur	239	115
Tin	455	235
Water, pure (ice)	32	0
Zinc	779	415

Colour and Temperature

	Fahr.	Cent.
Red, just visible	932	500
Dull red	1292	700
Cherry red	1472	800
Cherry red	1652	900
Orange	2184	1140
White	2408	1320
Dazzling	2732	1500

CHAPTER IV

FORMATION AND EXPANSION OF STEAM

Formation of Steam.—In a closed vessel, such as a boiler, containing partly air or steam and partly water, steam is produced by applying heat to the water. As the temperature rises, more and more vapour is given out from the water until at a certain temperature, called the boiling-point, ebullition begins. The steam as it is formed has only about the same volumetric space to fill, and gradually becomes compressed within the steam space above the water level; the pressure of the steam thus rises as more and more steam is generated. As the pressure rises the boiling-point also rises in conjunction with it, but at a certain pressure known as the working pressure, no further rise takes place because some of the steam is either conducted away through a pipe to the engine, or escapes through the safety valves which do not allow any further accumulation of pressure and limit it to a safe working load.

The steam escapes regularly or irregularly from the water surface according to the purity or impurity of the water. Only under very peculiar circumstances does each unit of surface yield an equal volume of steam. In an ordinary marine boiler steam is generally given off at some prominent point, perhaps a stay or rivet head, but the total area of the water surface plays its part in steam formation by supplying the heat necessary for evaporation and for breaking the cohesion of the molecules of water and converting them into steam. The evolution of the steam supplies the necessary circulation and movement of the water towards the points at which steam is given off.

In the process of raising steam and of afterwards maintaining a constant working pressure in a boiler, two changes must be considered —viz. the change of temperature necessary to bring the water up to the boiling-point, and the change of state necessary to convert the water into steam at the boiling-point.

Quality of Steam.—The steam formed in a boiler may be of two qualities, either *dry saturated* or *wet*. *Humid* or *wet* steam contains various proportions of moisture or particles of water in suspension.

When steam contains no moisture, and at the same time the temperature is not raised above that of the boiling-point due to its pressure, it is said to be saturated, or dry *saturated steam*. The term *saturated* means that the steam is saturated with heat because evaporation is complete.

In general practice, although both the temperature of the boiling-point and the pressure correspond to those of saturated steam (as shown on p. 18), there is some moisture in suspension, and the steam is humid or wet ; it is then called either *common steam*, if only a small proportion of moisture (3 to 5 per cent by weight) is present in it on leaving the boiler, or *surcharged* steam, if the proportion of moisture is large. *Priming* is said to take place when an undue proportion of moisture or water in suspension is carried off by the steam in its passage from the boiler.

When heat is added to saturated steam so that, all the moisture being then evaporated, the temperature of the steam is raised above that of the boiling-point due to its pressure, it is termed *superheated steam*. Steam cannot be superheated in the boiler in which it is generated under ordinary conditions of working, but if the water-level is allowed to fall below that of the heating surface, it is possible to superheat the steam ; but this is not a probable result or occurrence. When superheated steam is required the extra heat is added in a separate vessel to that in which the steam is formed, so that it shall not be in contact with the water from which it is generated.

In the ordinary process of producing superheated from common steam, the moisture is considered to be first converted into saturated steam (with no rise of temperature until the whole mass is saturated). As superheated steam, the temperature rises in a manner almost identical with a gas similarly heated. The specific heat of superheated steam near the boiling-point is about 0·48, and therefore for each b.t. unit of heat added to each 1 lb. by weight of dry steam a rise in temperature of about 2° Fahr. takes place in the superheated steam. Largely varying values have been found by experiment for the specific heat of steam at various pressures and degrees of superheat. It is highly probable that some moisture is always present, even when the steam is considerably superheated ; although this probability should not be overlooked, the generally accepted view, "that no moisture is present as soon as superheat begins to make its presence distinctly known" (by showing a temperature above that of the boiling-point due to its pressure), is sufficiently accurate for ordinary purposes.

The relative values of *pressure* and *temperature* of saturated steam are given in the Table of Properties of Saturated Steam at the end of Chap. II., which is based on experiment. The temperature rises with the pressure, but not in direct proportion as does a gas like air, and the tabulated results are nearly always used by engineers. Some empirical formulæ have, however, been derived from the table and used for instructional purposes, but they are of little practical use.

Total Heat of Formation of Steam.—From Regnault's and other experiments it is found that the latent heat of evaporation (see column 7, p. 18) decreases with an increase of pressure and temperature, according to the approximate formula—

$$\text{Latent heat, } L = 966 - 0{\cdot}7(T - 212),$$

where 966 = latent heat of evaporation of 1 lb. of water under atmospheric pressure; and T = the temperature of evaporation, or boiling-point.

If t = temperature of the water before heat is applied, then sensible heat, $S = T - t$ (see column 6, p. 18).

From the above, the *total heat of evaporation* of 1 lb. of water into saturated steam—

$$\begin{aligned} H &= L + S \\ &= 966 - 0{\cdot}7(T - 212) + (T - t). \end{aligned}$$

This formula shows, as well as the table, that the total heat of evaporation of saturated steam gradually increases, although the latent heat decreases, with the pressure and temperature.

Evaporation is generally considered to be *complete* evaporation of the liquid into gas. In the *formation* of steam some moisture is generally present in the steam, and evaporation is thus *incomplete*; and we thus speak of the "total heat of formation" of such steam, not of the "total heat of evaporation."

The proportion of dry steam, by weight, contained in each 1 lb., by weight, of common steam is called the *dryness fraction* of that steam. As stated above, there is always some moisture in common steam produced in a boiler; the whole of each 1 lb. by weight of water is, however, raised to the boiling-point, and the sensible heat $(T - t)$ remains unchanged; yet only a fraction or percentage of this 1 lb. may be converted into saturated steam. Let this fraction $= x =$ dryness fraction, and then $(1 - x)$ is the fraction by weight of moisture

present in the steam; then the latent heat expended in producing each 1 lb. of common steam is

$$= x \,.\, \mathrm{L}$$
$$= (x) \,.\, [966 - 0{\cdot}7(\mathrm{T} - 212)],$$

and the total heat of formation $= x \,.\, \mathrm{L} + \mathrm{S}$
$$= (x) \,.\, [966 - 0{\cdot}7(\mathrm{T} - 212)] + (\mathrm{T} - t).$$

It should always be remembered that when heat is taken away from the steam and it is reconverted into water, exactly the same quantity of heat is given up during the reconversion as was required originally to convert it into steam. Thus when steam is condensed to water (in the ordinary process of condensation, by adding water to it or by spraying the steam on to some cold surface), the steam gives up heat in exactly the same quantity as would be required to raise it from its state as water (to which it is reduced, and at the same temperature) to the temperature of the steam from which it is reduced, and to the same state of saturation, or condition of common steam.

EXAMPLE.—Find the quantity of heat given up by 1 lb. weight of steam at 295 lb. gauge pressure, and superheated to 620° Fahr., when reduced to ice at 0° Fahr. and normal atmospheric pressure.

		B.t. units.
Change of temperature,	sensible heat, from 620° to 420° = 200°, at 0·48 b.t. units per 1° F.	= 96
Change of state,	latent heat of evaporation, at 420° F.	= 815
Change of temperature,	sensible heat, from 420° to 32° F.	= 388
Change of state,	latent heat of fusion, at 32° F.	= 144
Change of temperature,	sensible heat, from 32° to 0° = 32°, at 0·5 b.t. units per 1° F.	= 16
	Total	= 1459

Some difference must exist between the pressure of the steam as it leaves the boiler and its pressure on arriving at the engine, or no motion or movement of the steam inside the pipe could take place. A fall in pressure must be accompanied by a fall in temperature. Again, there is another difference of pressure between that in the pipe near the engine, and that on the engine side of the valve which is used to regulate the supply of steam to the engine; at low powers this latter difference is very great, while the former is decreased. Whenever motion is produced, work is done and energy expended; consequently, some loss must take place in the internal heat energy of the steam in its passage through the steam pipe and through the regulating valve, producing a tendency to condense some of the steam to water, and therefore to produce a greater percentage of moisture in the steam on its arrival at the engine. This tendency to condensation is counteracted to some extent by the fall in pressure and temperature, which tends to release some of the heat necessary to keep the steam in a state of saturation or other original condition, because steam of lower pressure requires less total heat per 1 lb. than steam of the initial or higher pressure. If the steam leaves the boiler as saturated steam, seldom attained in practice, there is a slight tendency to produce a condition of superheating, because the steam may eventually possess a greater temperature than that naturally due to its pressure. The same remark applies to steam with little moisture; but in all cases, in addition to the losses of heat in doing work (in producing motion of the piston and of

the steam itself), there is a comparatively large loss by radiation of heat from the exteriors of pipes and valves, which prevents, or tends to prevent, any superheating of steam by a fall of pressure alone, in practice.

Expansion Curves.—Steam is not a perfect gas, and does not expand exactly in accordance with Boyle's law—P . V = constant—but somewhat differently, according to the conditions under which the engine is working. In Fig. 19 three curves are shown—

1. The *hyperbola*, which corresponds with Boyle's law.

2. The *saturation curve*, which is obtained by plotting a curve of pressures and corresponding volumes as shown in the Table for Saturated Steam. In this case, at any instant during the expansion, any increase of pressure will produce condensation of some of the steam, while any decrease of pressure tends to produce superheating :

$$P \cdot V^{1.0645} = \text{const.}$$

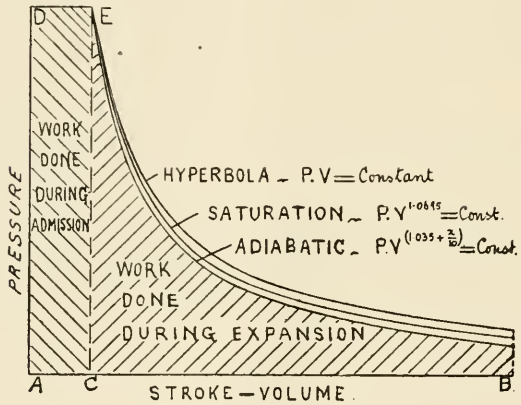


Fig. 19.—Expansion Curves.

3. An *adiabatic curve*, when steam expands in a cylinder without either taking in heat from, or giving out heat to, external bodies. Work is done by the steam in this case either by pushing a piston before it or by producing motion in itself, and thus creating kinetic energy :

$$P \cdot V^{1.035 + \frac{x}{10}} = \text{const.}$$

The above formula for adiabatic expansion of steam is attributable to Zeuner, and usually covers the range of moisture found in marine steam engine practice ; its limit of usefulness is reached when the moisture exceeds 30 per cent, *i.e.* when $x = 0{\cdot}7$.

If the steam be initially dry saturated, then $x = 1$,

and $$P \cdot V^{1.135} = \text{const.}$$

If the steam initially contains 20 per cent of moisture, then $x = 0{\cdot}8$.

and $$P \cdot V^{1.115} = \text{const.}$$

which may be written (approximately)

$$P \cdot V^{1\cdot 0} = \text{const.}$$

Hyperbolic Expansion of Steam.—For rough practical purposes the hyperbolic law of expansion holds good for steam as the working substance in the ordinary reciprocating (piston) engine; but, unlike the law as applied to gases, the expansion of steam is in all cases accompanied by a corresponding fall in temperature, and therefore the temperature is not constant.

For rough practical purposes, etc., steam, like all vapours, tends to expand infinitely, and on this property the economical working of the steam engine depends.

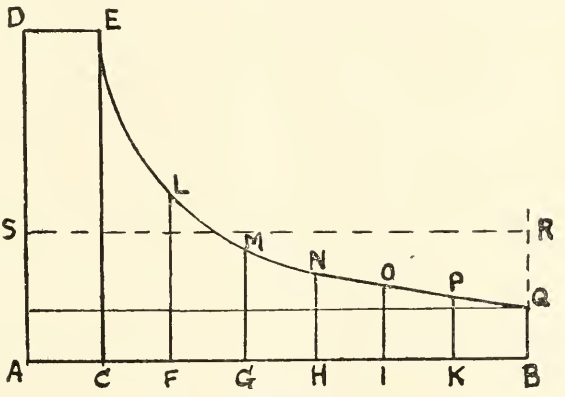


FIG. 20.—Hyperbolic Expansion.

In Fig. 20, let AB represent the stroke of the piston to any scale, and let AD represent the pressure in the cylinder at the commencement of the stroke. AD is at right angles to AB.

Suppose the steam to act with uniform pressure throughout AC, a fractional part of AB, and that at a point, E, steam is cut off, $CE = AD$, $DE = AC$; then at a point F, $CF = AC$, the pressure will fall to half that at E; represent this by $FL = \frac{1}{2}CE$ or $\frac{1}{2}AD$, and so on.

	lb.		lb.
Pressure at C is CE = say,	210	Mean, from A to C =	210·0
,, F is $FL = \frac{1}{2}CE$ =	105	,, ,, C to F =	157·5
,, G is $GM = \frac{1}{3}CE$ =	70	,, ,, F to G =	87·5
,, H is $HN = \frac{1}{4}CE$ =	52·5	,, ,, G to H =	61·25
,, I is $IO = \frac{1}{5}E$ =	42	,, ,, H to I =	47·25
,, K is $KP = \frac{1}{6}CE$ =	35	,, ,, I to K =	38·5
,, B is $BQ = \frac{1}{7}CE$ =	30	,, ,, K to B =	32·5
		Total =	634·50
		Mean of means, $\frac{1}{7}$ =	90·64

If a free curve be drawn through the points *E*, *L*, *M*, *N*, *O*, *P*, *Q*, this curve is a hyperbola, and it then represents the pressure at any point of the stroke graphically to scale.

In the figure, and in the calculation given above, the pressure is absolute pressure, of which the zero is represented by the line *AB*. If the exhaust pressure is zero absolute (and in good turbine practice it is only slightly above zero) the *work done* in the cylinder is represented to scale by the area *DENQBA*. But if the exhaust pressure line is above zero, the area contained between this line and the zero absolute line *AB* must be deducted from the area *DENQBA* to represent to scale the work done.

The work done on the piston may be divided into two parts—

1. The work done *during admission*, in which the steam pressure is retained at approximately its full or initial pressure, as represented by the area *ADEC*.

2. The work done *during expansion*, in which, after the supply is cut off and no more steam is admitted, work continues to be done, as represented by the area *CENQB*.

Figs. 11 and 19 show how *increased work* and economy of steam can be obtained from the same weight of steam by its *increased expansion*. By increasing the length of the cylinder and stroke of the piston the steam would continue to do work during its further expansion, although the amount admitted would remain the same. The reduction in pressure, obtained by increased expansion, is practically limited in piston engines to the pressure required to overcome the *back* pressure on the piston and the external resistance. There are also other practical limits exercised by considerations of weight and space.

When further admission of steam is cut off, say, at *C*; the ratio which *AB*, the full stroke, bears to *AC*, that part of the stroke before cut-off, is termed the *ratio of expansion* of the steam: it is generally denoted by *r*, and the cut-off then takes place at $\frac{1}{r}$ of the stroke. It is evident that the greater *r* is, the greater becomes the quantity of work which can be obtained from each 1 lb., by weight, of steam used in the cylinder; and thus steam is economised and greater efficiency obtained by increased expansion.

Economy by Increased Pressure is obtained by a possibility of increased expansion. Thus exhausting into the condenser, at 10 lb. absolute, the ratio of expansion with 150 lb. pressure is limited to $= \frac{150}{10} = 15$, and the ratio of expansion with 250 lb. pressure is limited

to $= \frac{250}{10} = 25$. The generation of steam of 250 lb. pressure costs very slightly more heat than the generation of steam at 150 lb. pressure, as shown by the table at the end of Chap. II., viz. as 1210·0 is to 1199·9 b.t. units.

Useful Work done by 1 lb. (weight) of Steam.—By constructing a figure, as shown in Fig. 19, to scale to represent graphically the absolute pressure P and the volume V of 1 lb. weight of steam, the quantity of work which it can do may be measured from the area of the figure. For purposes of comparison it is usual to assume zero absolute back or exhaust pressure, and to consider the cylinder, or piston, to be 1 square foot in area, so that its length up to the point of cut-off, or the stroke volume up to that point, is equal to the specific volume V of the steam at the absolute pressure P.

The work done during admission $= P \,.\, V$, and if the hyperbolic law holds good for steam, that is a *constant* quantity, and the work done by each 1 lb. weight of steam during admission is always the same whatever the pressure may be.

Similarly, the work done during expansion is also constant, if the same ratio of expansion be adopted, whatever the original pressure may be. But the drawback in practice is, that the lower pressure requires a larger cylinder in which to perform the work, varying directly with the absolute pressure.

For example, assuming hyperbolic expansion with a ratio of expansion of 4, an initial absolute pressure of 60 lb. and zero absolute back pressure, the mean pressure is 36·875 lb.

Under the same conditions but with 120 lb. initial absolute pressure, the mean pressure is 73·75 lb., or double that in the first case. But with the higher pressure the initial and final volumes of the steam are only one-half those in the first case, and consequently :
the *work done* in the first case

$$= 36{\cdot}875 \times 2 \,.\, V = 73{\cdot}75 \times V$$
$$= \text{work done in the second case.}$$

Equivalent Evaporation from and at 212° F.—The actual weight of water converted into steam by 1 lb. of fuel varies with the pressure and temperature at which it is formed, and with the temperature at which the feed water enters the boiler. Therefore, although the same quantity of heat may be usefully expended in making steam, the actual weight of steam produced would not be a fair measure for the comparison of evaporation per 1 lb. of fuel. A standard quantity

of heat, 966 b.t. units, the quantity necessary to convert 1 lb. of water into steam at 212° F. under normal atmospheric pressure, and when the feed water is at the same temperature, 212° F., is used by engineers for purposes of comparison. If the actual weight of steam produced per 1 lb. of fuel be n . lb., and the heat required per 1 lb. weight of steam $= x$. L + S, then the equivalent evaporation from and at 212° F.

$$= \frac{n \,.\, (x \,.\, \mathrm{L} + \mathrm{S})}{966}.$$

The expression, $\frac{x \,.\, \mathrm{L} + \mathrm{S}}{966}$, is generally called the "factor of evaporation." When the steam is dry saturated (*i.e.* when x = dryness fraction = 1)—

$$\text{equivalent evaporation from and at } 212° \text{ F.} = \frac{n \,.\, (\mathrm{L} + \mathrm{S})}{966}.$$

For *superheated steam*, taking the average value of the specific heat at 0·48 b.t. units, and the dryness fraction = 1, the

$$\text{equivalent evaporation} = \frac{n \,.\, (\mathrm{L} + \mathrm{S}) + (\mathrm{T}_s - \mathrm{T}) \times 0{\cdot}48}{996}$$

where T_s is the higher temperature of the superheated steam.

For values of specific heat see Figs. 132A and 132B, and for equivalent evaporation, from and at 212° F., see Fig. 136, Chapter XV.

CHAPTER V

MEASUREMENT OF POWER

Measurement of Pressure.—Work can be represented by an area, and this is also true of energy. Both work and energy are the products of two factors—*Force* in lb., and *Distance* in feet—which can be represented by the area of a plane figure, which is also the

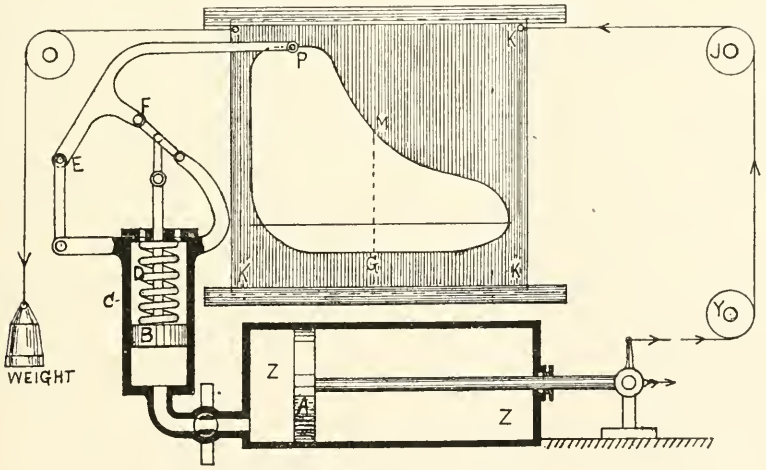


FIG. 21.—Principle of the Indicator.

product of two factors. This is practically used for measuring the work done in engines by graphical means with the assistance of the *indicator*.

A figure, such as is marked in Fig. 21, is obtained in practice by the use of the **Indicator** (more fully explained in a later chapter), whose general principle will now be examined and described.

In Fig. 21 there is a cylinder, *ZZ*, in which a piston, *A*, works. At one end of the cylinder there is a short pipe connecting it with the space below a small piston, *B*, which works vertically in a cylinder, *C*.

Above *B* there is a spring, *D*, which presses the piston downwards while the pressure below the piston pushes it upwards. The result of these two actions on the piston is that every inch of movement given to it, by change in pressure, represents a definite difference in the rise or fall of the pressure itself. To this piston, *B*, a rod and a connecting rod are attached, so that the movement of the piston also moves a lever, *EFP*, about its fulcrum *E*. A pencil, *P*, is attached to the lever at P, which can be pressed lightly on a piece of paper stretched on a board, *KKK*; thus as the pressure in the cylinder *ZZ* rises or falls, the pencil P also moves up or down, and makes a mark on the paper corresponding with the pressure to scale according to the proportions of the lever *EFP* and the strength of the indicator spring *D* which is in use.

The steam piston-rod has attached to it, in some simple way, a string, running over loose pulleys *Y* and *J*, which is attached to a board or flat piece of metal, *KK*, which can move easily to and fro in a slide. At the opposite end of *K* there is another loose pulley and string, and a weight which always keeps the string tight, so that when the piston *A* moves to the right then *K* moves to the left, and *vice versa*. To *KKK* the paper or indicator card is attached, and if the pencil is pressed lightly on the card while the piston makes two complete working strokes, one forward and one return stroke, a figure (shown) is marked by the pencil. This figure represents the pressure on the piston in the end of the cylinder connected with the indicator for the two complete strokes. On the forward stroke the pressure is represented by the upper bounding line of the figure, and on the return stroke by the lower bounding line, which is nearly horizontal. During the return stroke the back of the piston is only subject to exhaust or back pressure.

To obtain the **mean forward pressure,** a line must be drawn horizontally on the card, which must be to the proper scale to represent the atmospheric pressure. The top end of the indicator cylinder is always open to the atmosphere, and by connecting the bottom of the cylinder below the indicator piston with the atmospheric pressure (a small cock is fitted in the short pipe between *C* and *Z* for this purpose), the spring attains its natural length and is inoperative. Then by moving the card by hand or otherwise for the length of the stroke, a horizontal line representing the atmospheric pressure is marked by the pencil. By finding the mean height of the upper line of the figure above this atmospheric line we obtain the mean forward

pressure. Also by finding the mean height of the lower line above, or below, the atmospheric line we obtain the **mean back pressure** on the piston. From what has been stated above it will be seen that the—

$$\text{Mean } \textit{effective} \text{ pressure} = \begin{Bmatrix} \text{mean } \textit{forward} \\ \text{absolute pressure} \end{Bmatrix} - \begin{Bmatrix} \text{mean } \textit{back} \\ \text{absolute pressure.} \end{Bmatrix}$$

A similar diagram is obtained for the other end of the steam cylinder, and the mean pressure on the front of the piston can be obtained for the return stroke in a similar way.

Calculation of Mean Pressure.—The mean pressure is obtained from the indicator diagrams taken at opposite ends of the cylinder. There is a supposition that the figures obtained from the opposite ends are exactly similar, and the forward pressure at one end is usually taken in conjunction with the back pressure on the return stroke at the same end. The actual effective pressure on the piston at any point is, however, equal to the forward pressure *less* the back pressure on the opposite side of the piston. At a certain point, not far from the end of the stroke, the back pressure is greater than the forward pressure, and the inertia of the engine receives no effective assistance, but has to overcome a *negative* effective pressure.

Consider the pair of diagrams shown in Fig. 22.

The work done from left to right = area $ABKC$ – area MCE,
and from right to left = area $DEKF$ – area NFB.

Add together: then total work done per revolution, two strokes—

$$\begin{aligned} &= \text{areas } (ANFL + FKCL + BFK + NFB - MCE) + (DMCL \\ &\quad + FKCL + CKE - NFB + MCE) \\ &= \text{area } (ANFL + FKCL + CKE) + (DMCL + FKCL + BFK) \\ &= \text{area } ANFKECL + \text{area } DMCKBFL \\ &= \text{areas contained within the diagrams measured separately.} \end{aligned}$$

The *work done*, of which the indicator diagram gives an exact measure to scale, is represented by the area of the two figures contained within the lines drawn for each figure separately. Any variation of the exact similarity of the two figures produces a difference in the average turning-moment during the forward and return strokes. The examination of Fig. 116, an extreme case, shows such variation. Although the sum of the areas of the two figures represents accurately the work done in one revolution, the variation of turning-moment on the forward and return strokes is very great. In considering the

design of an engine, the calculation of turning-moments must be based on the actual difference of pressure on the two sides of the piston, not simply on the mean pressure obtainable from the diagram.

The diagrams are divided horizontally into ten equal parts by ten vertical lines (as shown in Fig. 22), the two outer parts forming one part, each being one-half. The heights contained between the upper and lower parts of each figure are measured for each stroke separately, and the mean height is calculated. The mean of the two means so obtained represents the mean pressure on the piston during one revolution—two strokes.

For convenience in measuring diagrams, a planimeter is used when a large number of diagrams are to be calculated ; but for ordinary

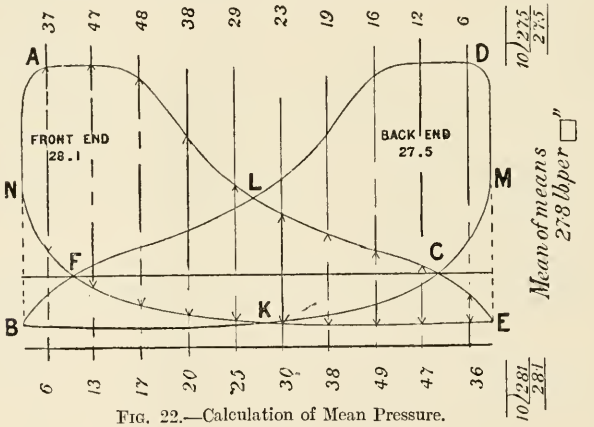


Fig. 22.—Calculation of Mean Pressure.

practice long slips of paper are run over the diagrams, and the heights for each stroke are ticked off in turn, from point to point, along one edge. This saves the labour of adding, and is less liable to error than taking each height separately. The total of these heights, divided by ten (the number of ordinates), gives the mean height and mean pressure to the scale on which the diagram is taken.

Indicated Horse-Power of an Engine.—A general idea of the horse-power of an engine is obtained by considering the resistance overcome by the steam when acting on the piston. A force equal to the difference in pressure (*i.e.* the effective pressure) on the two sides of the piston is used to overcome that resistance. The piston of an ordinary reciprocating engine moves with a varying velocity when rotating a crank, but if the crank makes N revolutions in each

minute, the piston then travels through 2N strokes, or twice the number of revolutions per minute. The force, therefore, acts through a distance, or creates motion by overcoming resistance through the same distance, per minute of 2NL feet, when L is the length of each stroke in feet.

The work done per minute is thus $= F \times 2NL$ foot-lb.

Now, as we have seen—

$$F = \left\{\begin{matrix}\text{the mean effective pressure per} \\ \text{square inch on the piston}\end{matrix}\right\} \times \left\{\begin{matrix}\text{area of the piston} \\ \text{in square inches}\end{matrix}\right\} = P \times A,$$

and work done per minute $= P \times A \times 2L \times N$ in foot-lb. per minute.

Also 1 H.P. = 33,000 foot-lb. per minute ;

therefore I.H.P. of engine $= \dfrac{2PLAN}{33{,}000}$.

The mean effective pressure on the piston is generally found in practice by using an indicator, and the power thus calculated by its use is called the Indicated Horse-Power (I.H.P). The I.H.P. developed in each cylinder of an engine is calculated separately, so that the total I.H.P. of the engine is the sum of the powers obtained in the several cylinders.

Calculation of Cylinder Constant.—As an example take the cylinder diameter at 27 inches, piston-rod diameter at $5\frac{1}{4}$ inches, and length of stroke at 33 inches. Find the cylinder constant.

The effective area on the forward stroke in the direct-acting engine is equal to the area of the piston, if there is no tail-rod ; on the return stroke the effective area is reduced by the area of the piston rod. For absolute accuracy the constants for the forward and return strokes should be measured separately, and multiplied by their respective mean pressures, and the mean taken of the two results thus obtained. In practice such extreme accuracy is unnecessary, and the mean of the two areas is taken as the value of A ; and the mean of the two mean pressures is taken as the value of P (in formula for calculating the I.H.P.).

In the example, therefore—

$$\begin{aligned} A &= \tfrac{1}{2}\{27^2 \times 0{\cdot}7854 + 27^2 \times 0{\cdot}7854 - (5\tfrac{1}{4})^2 \times 0{\cdot}7854\} \\ &= \tfrac{1}{2}\{572{\cdot}557 + 572{\cdot}557 - 21{\cdot}648\} \\ &= \tfrac{1}{2} \times 1123{\cdot}466 \\ &= 561{\cdot}733 \text{ square inches.} \end{aligned}$$

$$\text{The cylinder constant} = \frac{2 \cdot L \cdot A}{33{,}000} \quad (\text{and } L = \tfrac{33}{12})$$

$$= \frac{33}{12} \times \frac{1123{\cdot}466}{33{,}000}$$

$$= 0{\cdot}094622.$$

Revolution Constant.—For each engine, where diagrams are frequently taken, a table is generally compiled of revolution constants. Thus, this constant for the above example, at 110 revolutions, is—

$$0{\cdot}094622 \times 110 = 10{\cdot}48842.$$

It is now only necessary to multiply this constant by the mean pressure obtained at the corresponding number of revolutions to obtain the I.H.P.

For each cylinder of a stage-expansion engine a cylinder constant exists, and if the corresponding revolution constant be multiplied by the mean pressure indicated by the diagram, the product is equal to the power indicated in the cylinder. The sum of the powers thus obtained separately for each cylinder is the total horse-power indicated by the engine, or total I.H.P.

Brake Horse-Power.—The indicated horse-power is a measure

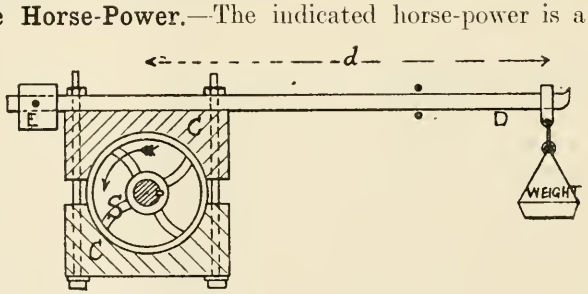


FIG. 23.—Prony Absorption Dynamometer.

of the work done per minute on the piston. Some of this power is used in overcoming the resistance of the engine itself to motion, such as frictional resistances of the piston in the cylinder, of the piston-rod packing, and of the bearings, so that only a certain proportion is actually utilised in rotating the shaft. This proportion is called the brake horse-power, and for small motors, whose energy can be entirely utilised for the test, is generally measured by means of a brake which absorbs the rotational energy of the shaft. (This form of brake is also called an *Absorption dynamometer.*)

In Fig. 23, *S* is a pulley on the shaft which is rotated by the engine.

Two nearly semicircular blocks of wood, *C*, *C*, are bolted together as shown about the pulley. To the upper block is secured a long lever *D*, and a short lever *E*, and at *E* a weight is placed so that *E* and *D*, and the apparatus generally, balance about the shaft centre, the bolts being then sufficiently slacked to allow free motion when the shaft is not in motion. A spring balance, or adjustable weight, can be suspended to the outer end *D* of the long lever.

Let R = the radius = $\frac{1}{2}$ diameter of the pulley ;
F = frictional resistance of pulley in the brake blocks, which must be tightened up as necessary.

Equating the moments of the forces acting about the centre of the shaft, the lever remains horizontal when $FR = Wd$, or when $F = \frac{Wd}{R}$.

The work done against the resistance (F) in one revolution of the pulley

$$= F \cdot 2\pi \cdot R \text{ foot-lb. ;}$$

and if the shaft turns N times per minute—

$$\text{the brake H.P.} = \frac{F \cdot 2\pi \cdot R \cdot N}{33{,}000} = \frac{W \cdot 2\pi \cdot d \cdot N}{33{,}000}.$$

In practice the engine is designed to run at a certain number of revolutions per minute, and is actually run at that speed when finding the B.H.P.

EXAMPLE.—In H.M.S. *Drake* a pull of 3000 lb. at a radius of 8 feet 9 inches is required to move the after-length of propeller shafting (friction test in dry dock) ; what H.P. is necessary to overcome the stern shaft friction when the shaft makes 120 revolutions per minute (full speed) ?

$$\text{H.P.} = \frac{W \times 2\pi \times d \times N}{33{,}000}$$
$$= \frac{3000 \times 2\pi \times 8\frac{3}{4} \times 120}{33{,}000}$$
$$= 600, \text{ about, for each of the twin shafts.}$$

When developing the full power of 30,000 I.H.P., about 1200 H.P. is thus absorbed, amounting to about 4 per cent of the full power. For a new ship the percentage would be much less ; but taking into consideration that the stern glands were temporarily unpacked, and special precautions taken to reduce friction by running water through the stern bearings, and that friction is also overcome in several plummer blocks not included in the test, the result is probably fairly accurate for working conditions, when 3000 lb. represents the *running*, and not the *starting*, pull.

Absorption dynamometers are only suitable for measuring low and moderate powers. They are commonly used for testing the H.P. (actually Brake H.P.) of engines used for driving motor cars, and the two figures usually quoted in the trade catalogues correspond to the B.H.P. of the axle shaft at two differing speeds which may be obtained under heavy and light loads. The B.H.P. can also be obtained by measuring the difference in tension of a belt or of a spring used in transmitting the energy of one shaft to another. A simple and fairly accurate arrangement of *belt transmission dynamometer* is shown in Fig. 24 (Thornycroft).

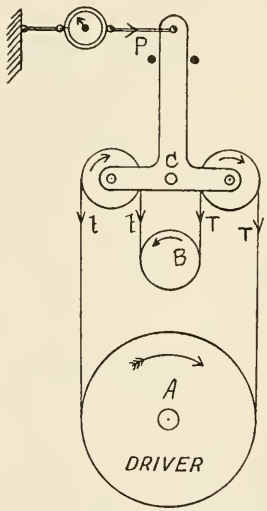


FIG. 24.—Thornycroft Transmission (belt) Dynamometer.

The belt is led over two loose pulleys, as shown, from the driver A to the driven shaft B. The loose pulleys are held by an inverted T-frame pivoted at C. The upper end of it is connected with a spring balance, P, or other simple device. When the shaft is revolved at its proper number of revolutions, neglecting friction of the pulleys, the pulls in the belt are respectively T and t; and the work transmitted per revolution $=(T-t)\times 2\pi\times R$ in foot-lb., where $R=$ radius of pulley B in feet.

Equating the turning-moments about C, and supposing the loose pulleys to be equidistant L from C, which is usually arranged—

$$2\,.\,T\,.\,L=2\,.\,t\,.\,L+P\,.\,CP$$

$$T-t=\tfrac{1}{2}P\,.\,\frac{CP}{L}.$$

Knowing the lengths CP and L and the radius R, and observing the pull P—

$$\left.\begin{array}{l}\text{The } \textit{brake horse-power} \text{ transmitted}\\ \text{at N revolutions per minute}\end{array}\right\}=\frac{P\,.\,\pi\,.\,R\,.\,N\,.\,CP}{33{,}000\times L}.$$

For engines driving dynamos, the *electrical* horse-power can be obtained by observing the ammeter and voltmeter; then—

$$\text{Electrical H.P.}=\frac{\text{ampères}\times\text{volts}}{746},$$

which is generally 90 to 95 per cent of the B.H.P.

$$\text{Electrical H.P.} = \tfrac{9}{10} \times \text{B.H.P.}$$
$$= \tfrac{9}{10} \times \tfrac{3}{4} \times \text{I.H.P.}$$
$$= \tfrac{1}{7}\tfrac{9}{4}\tfrac{9}{0} \times \text{I.H.P.}$$

The Walker air brake is very useful for experimental purposes, and for tests in the shop the Froude water brake is frequently used, especially for motors and aeroplane engines.

Shaft Horse-Power.—For the measurement of power there are many methods, all more or less elaborate. Generally speaking, fairly accurate results are obtainable with an indicator suitably adapted to the special type of engine under test. It is quite out of the question to assume that any particular indicator is suitable to all cases, and especially is this true of the particular case of marine motor-engines running at high rates of revolution and with great range of pressure within the cylinder. With absorption dynamometers of the Prony brake type the whole of the shaft-power is absorbed by the brake, and therefore they are useless for determining the power transmitted from an engine to some useful purpose.

When transmitting power a propeller-shaft withstands a stress, compounded of the actual twist imparted to it, and of the propulsive thrust-compression which, according to Dr. Hamilton Gibson's experiments, may be as much as 18 per cent of the stress due to torque alone. The torque is the turning-moment in foot-lb. required to produce a corresponding angular twist, and the *torsion-meter* is used to measure this angular twist between two points on a shaft. For a correct scale of measurement the shaft between these two points must be of uniform external and internal diameter, and of homogeneous material.

In the *Hopkinson-Thring Torsion-meter*, a sleeve, AC, whose length is generally 4 or 5 times the diameter of the shaft, is fixed to the shaft at one end, A, and terminates in a collar, C, opposite to another collar, B, fixed to the shaft. Any angular twist of the shaft between the two points A and B is thus shown by the relative position of C and B, in connection with each of which a set of mirrors reflects a ray of light from a lamp on to a scale, as shown in Fig. 24A. The rays, even at moderate speed, appear as continuous lines on the scale, and the distance between these two lines represents to scale the angular twist of the shaft in the length AB. The " zero " reading is obtained when the shaft is rotated under no load, either by disconnecting the propeller, or by allowing the shaft to be dragged round by the propeller for a few moments by the way on the vessel, when the two rays of light

should correspond identically with each other, and, for convenience of direct reading when transmitting a load, with the zero on the scale.

Dr. Gibson made experiments with material to obtain the elastic qualities. A torque was applied on a shaft by means of two spring balances at opposite ends of a cross-bar fixed on the end of a shaft and forming a couple which tended to eliminate friction in the supporting bearing. At the initial or dead point of measurement the shaft

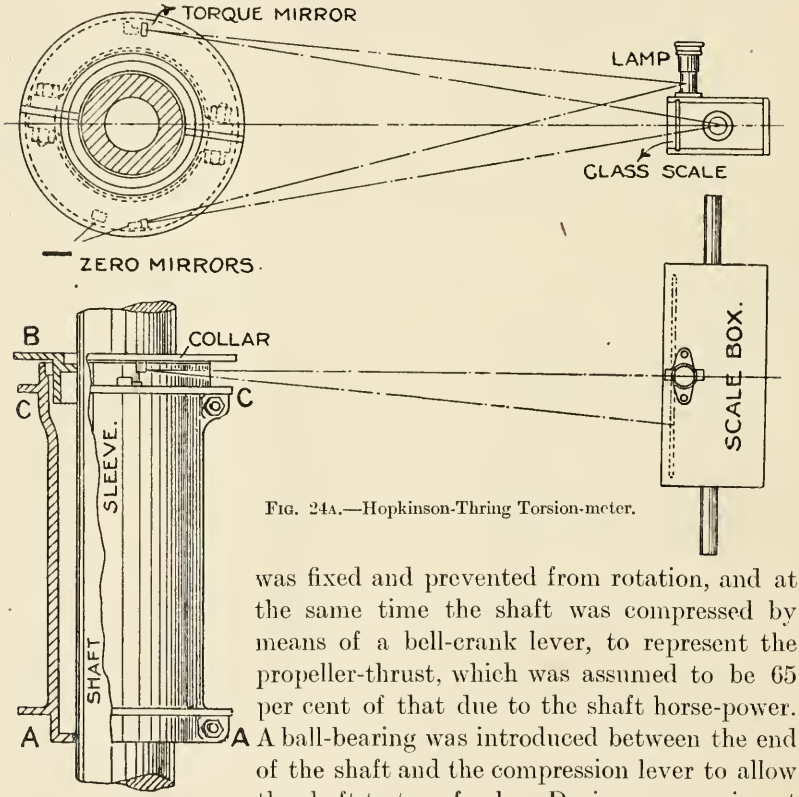


FIG. 24A.—Hopkinson-Thring Torsion-meter.

was fixed and prevented from rotation, and at the same time the shaft was compressed by means of a bell-crank lever, to represent the propeller-thrust, which was assumed to be 65 per cent of that due to the shaft horse-power. A ball-bearing was introduced between the end of the shaft and the compression lever to allow the shaft to turn freely. During an experiment of this kind it is necessary to keep the shaft "alive" by tapping its surface with lead hammers because there may be some tendency to "lag" when putting the load on and off.

Fig. 25 shows an outline profile of the *Chadburn Flash-light Torsion-meter* (Bevis and Gibson's patent) applied to a shaft with three plans under various conditions, and the general construction of the apparatus for use with turbine engines and with reciprocating engines respectively.

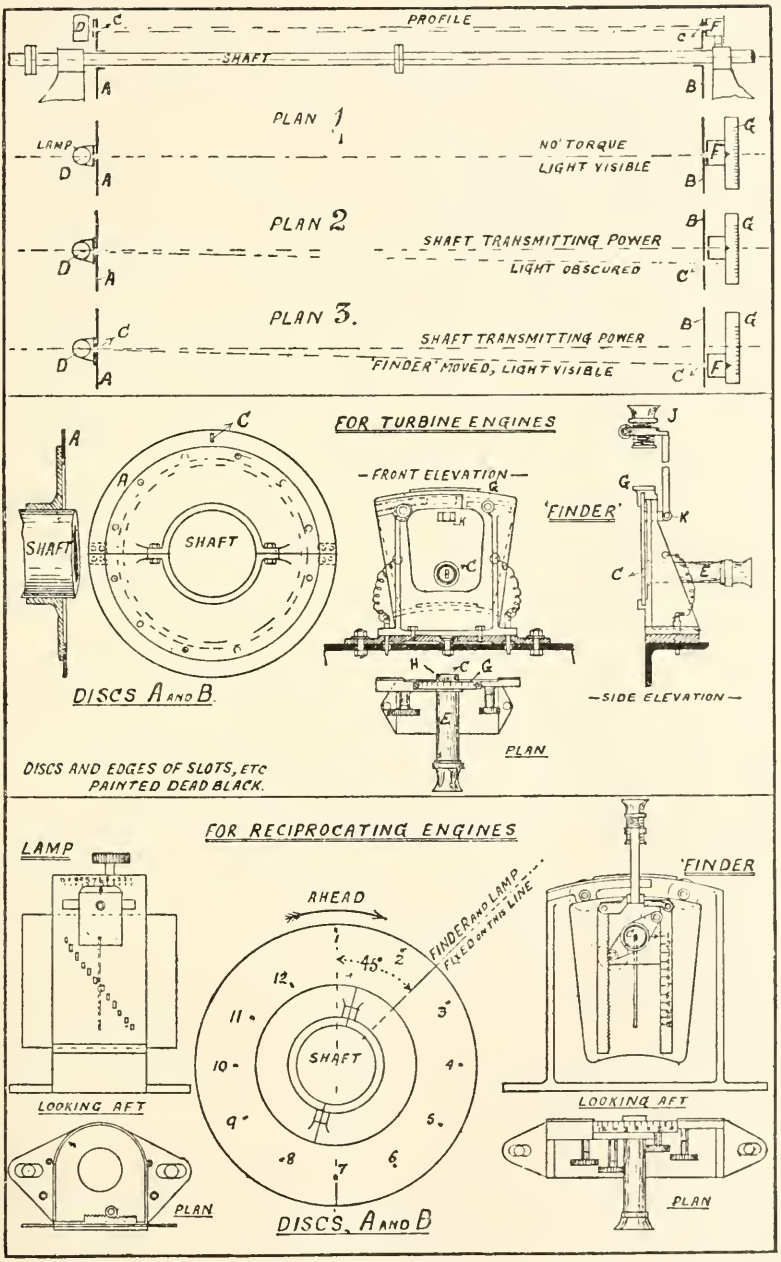


FIG. 23.—Flash-light Torsion-meter.

Each apparatus consists of two metal discs, A and B, rigidly secured to the shaft, and for *turbine engines* each disc has a single small radial slot, C, near the periphery, accurately adjusted so as to lie exactly in the same radial plane containing the axis of the shaft. The discs and edges of the slots are painted a dead black so as to reflect as little as possible, and thus not confuse the single beam from the lamp. The portable lamp, D, is accurately secured behind one disc, A, so that the light shows through both discs when the slots are in opposition. The "finder," F, is also portable, but so made as to be readily mounted behind the other disc, B, and secured quickly and accurately in position. A movable eye-piece, E, is fitted to the finder, F, and E is adjustable circumferentially by means of a delicate micrometer screw-gear which sets the finder with extreme accuracy. There is a machine-divided scale, G, and a vernier, H, on the finder, which is also provided with a magnifying reader, J, to enable the scale to be read off to $\frac{1}{100}$th of a degree, or $\frac{1}{360000}$th of the circumference of the circle described by the slots when the discs revolve. The torque-finder is the most delicate part of the apparatus, and when not in use should be unshipped and carefully stowed in its box. The discs should also be suitably guarded by casings if left in place.

When transmitting a load the torque-finder is moved until the exact angle of the twist of the shaft allows the flash to be clearly seen through the eye-piece. For a rapidly revolving shaft, such as the propeller-shaft of a turbine steamer, the light as seen through the torque-finder is practically continuous; at lower speeds the light appears to flicker, but there is no difference either in the accuracy of the reading or in the ease with which the torque is obtained. Typical of this accuracy may be cited the following experience with a Channel turbine steamer:—Running at full power, about 2600 horse-power on one of three shafts, and at 470 revolutions per minute, torsion-meter records were taken, as well as at various progressive speeds down to 24·4 R.P.M., when only 9·2 shaft horse-power was developed. During the trials of a 33-knot destroyer running at about 750 revolutions per minute, one observer was able to get the horse-power of the three separate shafts while running each measured mile, *i.e.* in less than two minutes.

In cases where a long length of shaft is not available, a modification in the disposition of the apparatus is made, and the flash is thrown out radial to the shaft instead of in the direction of its axis. This arrangement involves more material in the apparatus, and it is some-

what more difficult to fit up; but the accuracy is maintained, and readings can be well obtained from a piece of shaft not more than two or three diameters long.

For the shaft of *reciprocating engines*, or any shaft exposed to a fluctuating torque, means are provided to enable the observer to take twelve or more separate readings for one revolution. For this purpose the light and the eye-piece are moved radially to or from the shaft so as to come opposite each pair of slots in the discs, and observations made for each position. When these torque readings are plotted out on a development of the crank circle, as in a twisting-moment diagram, the resultant curve is found to agree very closely with the diagram built up from the corresponding set of indicator-diagrams taken from the engine cylinders. The difference between the two curves shows, with remarkable fidelity, the power absorbed by engine friction (see Fig. 131).

The calculation of the shaft horse-power is based on Rankine's formula, and for this purpose it is necessary to know or to observe—

A = angle of twist of shaft, in degrees;
N = number of revolutions per minute of the shaft;
D = external diameter of the shaft, in inches;
d = internal diameter of the shaft, in inches;
(when the shaft is solid, $d = 0$)
L = length of shaft between the points of measurement, in inches;
(if a coupling is included its length should be deducted)
K = constant dependent on the modulus of elasticity;
= 3·27 for a steel shaft with a modulus of 11,250,000.

Then—

$$\text{Shaft H.P.} = \frac{A \cdot N \cdot (D^4 - d^4)}{K \cdot L}.$$

For the same shaft and measuring apparatus, the values of D, d, K and L do not change, so that—

$$\text{Shaft H.P.} = C \cdot A \cdot N$$

where C = a constant for the particular shaft

$$= \frac{D^4 - d^4}{K \cdot L}$$

= shaft constant.

The calculation of the shaft horse-power thus resolves itself into a

simple sum in multiplication; and for each ship, by tabulating the values of C . N (the revolution constant) the calculation can be still further simplified.

Efficiency.—In every mechanism or engine there is a certain wasteful expenditure of energy, so that the useful work done is always less than the energy expended. The efficiency of a machine is represented by a ratio, fraction, or percentage—

$$\text{Efficiency} = \frac{\text{useful work done (in ft.-lb.)}}{\text{energy expended (in ft.-lb.)}}.$$

In marine propulsion there are *five* elements, each of which absorbs a certain proportion of the energy expended. These elements are: the *boiler*; the working substance, *steam*; the engine *mechanism*; the *transmission* gearing and *thrust* mechanism; and the *propeller*. Finally, only a very small proportion of the energy of the fuel is expended in usefully urging the ship along. A few typical examples, estimated on such details as are available, show the *total* or *overall efficiencies* of several modern marine propelling plant, working *at full power*. The overall efficiency is the product of the efficiencies of the five elements, and, in the table below, each efficiency is shown as a percentage.

	Boiler.	Steam.	Engine.	Thrust and Transmission Gearing.	Propeller.	Overall.
Reciprocating, *Topaze*	53 ×	11 ×	90 ×	85 ×	65 =	3 per cent
Turbines, *Amethyst* (direct driven)	52 ×	16 ×	95 ×	95 ×	55 =	4·25 ,,
Turbines, *Dreadnought* (direct driven)	65 ×	17 ×	98 ×	98 ×	58 =	6·21 ,,
Reciprocating, *Caronia*	70 ×	16 ×	90 ×	90 ×	69 =	6·26 ,,
Turbines, *Carmania* (direct driven)	70 ×	17 ×	98 ×	98 ×	60 =	6·82 ,,
Turbines, single reduction (warships)	70 ×	18 ×	98 ×	97 ×	65 =	7·78 ,,
Turbines, double reduction (merchant vessel)	70 ×	19 ×	98 ×	95½ ×	65 =	8·12 ,,
Diesel Oil Engine (fuel)		32 ×	80 ×	98 ×	60 =	14·05 ,,

At lower powers generally, the efficiencies of the boiler and propeller increase for both reciprocating and turbine propelled vessels; the efficiency of the steam generally increases for reciprocating and decreases for turbine engines; the efficiencies of the engine and thrust decrease for both reciprocating and turbine engines. The various efficiencies are explained briefly below, and further information is given in fuller detail later on in the book.

1. *The Boiler.*—The heat energy contained in the fuel consumed

is not all transferred to the steam generated in the boiler. There are losses from imperfect combustion; from heat passing away by radiation, conduction, and other causes, to the stokehold and its surroundings; and by the passage of the hot gases through the funnel into the atmosphere. Finally, instead of 15 lb. of water being converted into steam, from and at 212° F., by burning 1 lb. of coal, only 10 to 11 lb. weight of steam leave the boiler, and thus—

$$\text{Efficiency of boiler} = \frac{10 \text{ to } 11}{15} = 0{\cdot}66 \text{ to } 0{\cdot}73 = 66 \text{ to } 73 \text{ per cent.}$$

When the fires are urged by excessive forced-draught, the efficiency is low, as shown above by the *Topaze* and *Amethyst*; at lower powers the efficiency becomes 65 for *Amethyst* and 62 per cent for *Topaze*. This comparatively low efficiency is attributable to the closed stokehold system and the absence of heat-saving appliances, such as hot-air draught and feed heaters or economisers. The *Caronia* and *Carmania* have tank boilers fitted with Howden's closed ashpits and a system of heating the air, by the waste funnel gases, before it is supplied to the fires.

2. *The Steam.*—Of the total heat energy leaving the boiler in the form of steam there is a moderate loss by radiation, etc., and the exhaust enters the condenser principally as steam, but partly as water. The heat in the exhaust steam and water, commonly known as the *exhaust waste*, or "heat rejected," is absorbed by the condenser circulating-water and pumped into the sea. To a very large extent the exhaust waste is unavoidable, therefore—

$$\text{Efficiency of the steam} = \tfrac{1}{10} \text{ to } \tfrac{1}{5} = 0{\cdot}1 \text{ to } 0{\cdot}2 = 10 \text{ to } 20 \text{ per cent.}$$

Turbine engines give a greater efficiency than reciprocating engines at full power; but at powers below about one-half, the reciprocating engine is more efficient, unless cruising systems are fitted, when the power may be reduced to about one-sixth the full power with about an equal efficiency.

3. *The Mechanism of the Engine.*—The brake horse-power or shaft horse-power is measured at the part of the shaft between the engine coupling and the transmission gearing or thrust-block, the

$$\text{Efficiency of the mechanism} = \frac{\text{brake horse-power}}{\text{indicated horse-power}}.$$

Of the work done on the pistons (actually measured by the use of the indicator and known as indicated work), some is absorbed in overcoming frictional resistances of the slide valves, pistons, packings, guides, bearings and other working parts, and thus only about 75 per cent at low powers, up to 94 per cent at full power, is transferred into useful work in rotating the shaft—

Efficiency of the mechanism = 0·75 to 0·94 = 75 to 94 per cent.

The frictional resistances of the engines of H.M.S. *Hyacinth* and *Minerva*, without any load (propellers removed, and therefore also without thrust resistance) and at full speed, were found to absorb 5 per cent of the full I.H.P. Working under a load the frictional resistances must necessarily be greater, because there is an increased pressure on all bearing surfaces, packing, etc., together with an increased temperature of all internal working parts.

By using a Fottinger flash-light apparatus for measuring the shaft H.P. of the *Agamemnon*, and the German cruiser *Hamburg*, it was found that 7 and 6 per cent of the I.H.P. respectively was absorbed in engine friction at full power; but at half-power the mechanical efficiency was reduced to about 82 or 80 per cent. Roughly, at low powers of reciprocating engines the mechanical efficiency

$$= E\sqrt{\frac{n}{N}},$$

where E = efficiency at full power;
N = revs. per minute at full power;
n = revs. per minute at the lower power.

With turbine engines the internal friction is less than with reciprocating engines, and the mechanical efficiency is seldom less than 95 per cent, and at full power with recently constructed turbines and forced lubrication it may be taken as 98 per cent.

4. *The Transmission Gearing and Thrust Mechanism.*—What may best be described as the engine mechanism is complete in itself up to and including the crank or shaft coupling; beyond this point any means of transmission may be interposed to give effect to any other element or mechanism.

In marine engines with a direct drive conveyed to the propeller by a shaft coupled directly to the propeller shaft, the thrust-block mechanism is the first and principal consideration. If gearing is

fitted, either of the single or double reduction type, it is necessarily interposed between the crank or shaft coupling and the thrust mechanism (see Chapter XXII.).

Friction in the thrust bearing may be considered to increase directly as the pressure on the thrust collars is increased in urging the ship along, or as about the *square of the speed* of the ship (see Chapter XXVII.).

From experiments made with reciprocating engines and ordinary multicollar thrust-blocks in good class practice, it has been found that thrust friction absorbs from 6 to 7 per cent of the power at full speed, and probably over 10 per cent at reduced speeds and lower powers. With indifferent construction and lubrication it may amount to 20 per cent of the I.H.P.

The Michell thrust-block reduces the frictional loss in ordinary cases to one-fifteenth or one-twentieth of that obtaining with the older types of multicollar blocks of good construction, and the loss of efficiency in the thrust mechanism may thus be reduced to less than 1 per cent, and a general average efficiency of 98 per cent may be usually assumed for the Michell single collar thrust with its contingent film lubrication.

With direct driven turbines the whole of the propeller thrust is nearly absorbed by the difference of pressure within the turbines themselves without any expenditure of energy, and an efficiency of 98 per cent may be assumed.

With transmission gearing the slight difference of thrust within the turbine is balanced by the turbine thrust-block, and any loss due to this cause is considered as part of the turbine engine mechanism, under (3) above.

The whole of the propeller thrust must be taken independently of the transmission gearing, and an efficiency of about $98\frac{1}{2}$ per cent may be assumed when the Michell thrust is fitted, which is now the common practice.

Double helical gearing of the single reduction type has an efficiency of about $98\frac{1}{2}$ per cent. If two sets of double helical gearing are used and combined so as to obtain double reduction, there is a loss of about $1\frac{1}{2}$ per cent in each set, and the overall efficiency of double reduction gearing is thus about $98\frac{1}{2} \times 98\frac{1}{2} = 97$ per cent.

At **full power,** the overall efficiency of the transmission gearing and the thrust mechanism is thus about—

For reciprocating engines, direct drive and multicollar thrust-blocks . .	80 to 90 per cent
For turbines, direct drive with turbine thrusts	95 to 98 ,,
For turbines, single reduction gearing and Michell thrust	$98\frac{1}{2} \times 98\frac{1}{2} = 97$,,
For turbines, double reduction gearing and Michell thrust	$97 \times 98\frac{1}{2} = 95\frac{1}{2}$,,

5. *The Propeller.*—Energy is lost by screw propellers from the oblique action of the blades, only a moderate proportion of the water acted on being moved in a direction opposite to that of the ship, and the remainder is driven away in other directions which do not assist the ship along; other disturbances, such as eddies, are also caused in the water passing between the blades. There is considerable friction between the water and the blade surfaces, which tends to increase disturbance and wasteful effect of oblique action; and there is also cavitation in some cases (see Chapter XXVIII.).

Screw propellers directly driven by turbines are rotated at a greater number of revolutions per minute than those driven by reciprocating engines, and consequently there is greater frictional resistance to the passage of the blades through the water and correspondingly greater wasteful disturbance.

Single reduction gearing, now frequently fitted in connection with turbine machinery, allows the propeller to run at a more efficient rate of revolution, and consequently the propeller efficiency is increased from about 42 per cent to about 50 per cent in destroyers up to 60 per cent for larger vessels.

Double reduction gearing enables a further increase to about 55 per cent in small craft, and in mercantile vessels the efficiency may be assumed to be about 60 to 65 per cent, thus corresponding to that attainable with slow speed reciprocating engines in cargo-carrying steamers.

At **full power**, the efficiency of the propeller may be—

For reciprocating engines of slow speed, or for turbines with double reduction gear .	60 to 65 per cent
For turbines with single reduction gearing .	55 to 60 ,,
For turbines with direct drive . . .	45 to 55 ,,

The *effective or propulsive horse-power*, which is the product of the efficiencies of the engine mechanism, transmission gearing and thrust

mechanism and the propeller, is seldom greater than 60 per cent at full power of the I.H.P. of reciprocating engines or the S.H.P. of turbines (see Chapter XXVII.). Generally the highest efficiency of the effective power is obtained at the full designed power, and there is a gradual decrease in efficiency as the power decreases, but as it is now practicable to accurately measure the effective thrust on the propeller by means of the adaptation of the Michell thrust to this purpose, as described later, much more light will be thrown on this subject which has been hitherto obscured by somewhat theoretical calculations of an abstruse character. Results have, however, generally been closely approximate to these calculations, and show that the subject has received careful analysis.

Maximum Efficiency of a Perfect Heat Engine.—In a heat engine using a perfect gas as the working substance, the quantity of heat, Q, contained in unit mass of the gas is directly proportional to its absolute temperature, T_0. Only a certain portion of the total quantity of heat in the substance can be changed into useful work, because there is a certain absolute temperature, t_0, below which the engine cannot be worked. The mass of gas, therefore, leaves the engine with a quantity, q, of heat in it, which corresponds evidently to an absolute temperature, t_0.

The heat which can be utilised $= Q - q$, and the efficiency then obtained becomes—

$$= \frac{\text{heat usefully expended}}{\text{total heat available}} = \frac{Q - q}{Q} = 1 - \frac{q}{Q},$$

in which $\frac{q}{Q} = \frac{t_0}{T_0}$.

Therefore efficiency $= 1 - \frac{t_0}{T_0} = \frac{T_0 - t_0}{T_0}$, in *absolute* temperatures;

or $$= \frac{T - t}{T + 461},$$

where T and t are measured from the ordinary zero, 0° Fahr.

This ratio $\frac{T - t}{T + 461}$ is termed the maximum efficiency of a perfect heat engine, and is frequently used for comparison of heat engines, one with another. This maximum efficiency can never be actually realised in practice although applicable to the steam engine as a heat engine. The actual quantity of heat contained in a mass of steam,

either on entry to or exhaust from the steam engine, is dependent on the dryness fractions of the steam, both of which may vary considerably, although the temperatures, T and t, may remain constant. This variation from actual truth does not, however, detract from its usefulness in the general comparison of heat engines, whether using the same or some different working substance, and the following tabular statement of the maximum possible steam (heat) efficiencies under various conditions is instructive :—

		Possible increase per cent.
For Newcomen's engine	$\frac{212-150}{212+461} = \frac{62}{673} = 0{\cdot}092$ about.	
For a non-condensing engine, 50 lb. pressure	$\frac{297-212}{297+461} = \frac{85}{758} = 0{\cdot}112$ about.	22
For a condensing engine, 50 lb. pressure	$\frac{297-100}{297+461} = \frac{197}{758} = 0{\cdot}26$ nearly.	132
For a condensing engine, 100 lb. pressure	$\frac{338-100}{338+461} = \frac{238}{799} = 0{\cdot}298$ nearly.	14·6
For a condensing engine, 150 lb. pressure	$\frac{366-100}{366+461} = \frac{266}{827} = 0{\cdot}321$ about.	8
For a condensing engine, 200 lb. pressure	$\frac{388-100}{388+461} = \frac{288}{849} = 0{\cdot}34$ nearly.	$6\frac{1}{4}$
For a condensing engine, 250 lb. pressure	$\frac{406-100}{406+461} = \frac{306}{867} = 0{\cdot}353$ about.	3·8
For a condensing engine, 300 lb. pressure	$\frac{422-100}{422+461} = \frac{322}{885} = 0{\cdot}364$ about.	3·1
For a condensing engine, steam super-heated to 639° F.	$\frac{639-100}{639+461} = \frac{539}{1100} = 0{\cdot}49$	34·6

Summary of Formulæ

966 = Latent heat of 1 lb. saturated steam under atmospheric pressure and a temperature of 212° F.

$L = 966 - 0{\cdot}7\ (T - 212)$

$S = T - t$

$H = L + S$

$= 966 - 0{\cdot}7\ (T - 212) + (T - t)$, for saturated steam

$= x\ .\ L + S$, for wet steam

$= x\ .\ \{966 - 0{\cdot}7\ (T - 212)\} + (T - t)$

1 British Thermal unit = 780 ft.-lb. = J = Joule's equivalent. Expansion curves—

$$\text{Hyperbola, } P \cdot V = \text{constant.}$$
$$\text{Saturation, } P \cdot V^{1\cdot0645} = \text{constant.}$$
$$\text{Adiabatic, } P \cdot V^{1\cdot035+\frac{x}{10}} = \text{constant.}$$

Equivalent evaporation, from and at 212° F.—

$$= \frac{n \cdot (x\mathrm{L} + \mathrm{S})}{966}.$$

Maximum efficiency of a heat engine using a perfect gas as the working substance—

$$= \frac{\mathrm{T} - t}{\mathrm{T} + 461}$$

$$\text{I.H.P.} = \frac{2 \cdot \mathrm{P} \cdot \mathrm{L} \cdot \mathrm{A} \cdot \mathrm{N}}{33{,}000}, \text{ for a double acting reciprocating engine.}$$

$$\text{Brake H.P.} = \frac{\mathrm{W} \cdot 2\pi \cdot d \cdot \mathrm{N}}{33{,}000}$$

$$\text{Shaft H.P.} = \frac{\mathrm{A} \cdot \mathrm{N} \cdot \mathrm{D}^4}{\mathrm{K} \cdot \mathrm{L}} \text{ for a solid shaft}$$

$$= \frac{\mathrm{A} \cdot \mathrm{N} \cdot (\mathrm{D}^4 - d^4)}{\mathrm{K} \cdot \mathrm{L}} \text{ for a hollow shaft.}$$

PART II

CHAPTER VI

MARINE BOILERS—WATER-TANK TYPE

The marine boiler of the present day is essentially a high-pressure boiler, and to a large extent it depends for its power of converting water into steam on the area of tube surface which it contains. The boiler is called a *tank* boiler if the gases pass through the tubes on their way to the uptake and funnel, or chimney; this boiler is usually *internally* fired, and the furnaces and the tubes are surrounded by water. If the gases pass around and between the tubes, the boiler is *externally* fired, and the tubes contain water and sometimes partly steam; the boiler is then of the *water-tube* type.

In this chapter the boilers described are of the tank type, and consist principally of cylindrical boilers with nearly flat ends. They are as follows:

Return tube cylindrical boiler, single ended.
,, ,, ,, double ended.
Direct tube cylindrical boiler, now seldom used; and
,, ,, locomotive boiler, as used on railways, etc., but abandoned for marine use in favour of upright water-tube boilers.

The various types of water-tube boilers may be divided into two fairly well defined classes, namely, those in which the—

1. Tubes are nearly *upright*, and in consequence of which the natural circulation is very rapid, and the tubes are of small diameter. (Chap. VII.)

2. Tubes are nearly *horizontal*, and in consequence of which the natural circulation is comparatively slow, and the tubes are necessarily

of large diameter to prevent overheating of the material. (Chap. VIII.)

At the end of Chap. VIII. there are some notes on the "Influence of Practice on Boiler Construction and Design" which are of a somewhat more advanced character than the descriptions of the various types of boilers. The notes, however, shed considerable light on the direction of future construction.

Historical Introduction.—Before and during the early Newcomen period the ordinary boiler was naturally something of the shape of a kettle or covered saucepan. Newcomen adopted an almost globular shape, and this continued to be used for many years. It must be remembered that in those days iron or steel plates of uniform thickness were almost unknown, and that the plating of the early boilers had to be hand forged from the pig-iron blocks. Some of the smaller boilers were constructed of two dished plates as shown in Fig. 4.

With the demand for greater powers the boiler itself became a larger affair, and a boiler, very similar in shape to a *haystack*, but built of many plates riveted together, became a common form. At the beginning of the nineteenth century the boiler developed into a cylindrical shape with rounded ends, but a more general type was *box* shaped; both these boilers were *internally* fired, that is, flues or furnaces were introduced into the interior of the boiler, and, generally, each flue was surrounded by water. The "*Galloway*" *boiler* of the present day is a survival, in an improved form, of the early cylindrical round-ended boiler. The pressure in the early boilers seldom exceeded 5 or 10 lb. per square inch, although Watt, who clearly understood the advantage of higher pressures, attempted to obtain a boiler for 50 or 60 lb. working pressure.

No great advance was made on the *flue boiler* until about 1840, when the *Penelope* frigate was fitted with boilers of box shape, containing a large number of tubes, and called *multitubular boilers*. With the exception of improvements in material and design the multitubular box boiler was in general use in the Royal Navy until 1870; but these improvements allowed an increase in pressure from 5 or 10 lb. up to 35 or 40 lb. per square inch. In tubular boilers, the gases and flames from the furnaces pass through the tubes on their way to the uptake and funnel, and the tubes are generally horizontal and surrounded by water.

Cochran's boiler took the form of a mixed flue and water-tube boiler: the tubes were vertical and filled with water, and the gases

surrounded the tubes on their passage to the uptake. This boiler gave an increased efficiency for a short period ; but the smoke passages could not be properly swept while under steam (a defect shared by many water-tube boilers more or less well known at the present day), and the efficiency fell in a few hours to something below that of the ordinary multitubular type.

From the multitubular box-boiler grew the *cylindrical marine boiler* of the present time. Some boilers (called *oval*) were flattened at the sides with a semicircular top and bottom. The pressure rose in some instances to 60 lb. per square inch with the introduction, about 1860, of the surface condenser, which contributed very largely to the possibility of using increased pressures by decreasing the frequency of the necessary scaling and cleaning. With higher pressures came the introduction of compound or stage-expansion engines, and the *Alexandra* (1871) was one of the first naval vessels to be fitted with cylindrical boilers and compound engines working at 60 lb. pressure. A slight increase of pressure followed, but was adopted slowly, until the introduction about 1880 of the evaporator solved the difficulty of scale forming in the boilers. From this time the growth in pressure was rapid, and in 1900 was commonly 300 lb. for naval boilers of the water-tube type, and 200 to 210 lb. for cylindrical boilers. Beyond these pressures there has been no general advance except in a few particular instances.

The general introduction of water-tube boilers took place about 1893–94, and was brought about by a desire to obtain a greater power with safety on less weight—an object which has been very distinctly attained in all modern naval vessels, the total weight of boilers and machinery in which is at most one-half that of mercantile vessels. The marine cylindrical boiler will produce continuously and safely from 15 to 18 I.H.P. per square foot of grate surface, equivalent to a coal consumption of $22\frac{1}{2}$ to 27 lb. of coal per square foot of grate. Any attempt to increase the consumption beyond this amount is likely to produce permanent defects in the boiler. The limit is very clearly defined for boilers of the usual cylindrical construction. With naval water-tube boilers a far greater consumption can be safely obtained, varying from 30 to 100 lb. per square foot of grate area, corresponding to the type of boiler and the angle of inclination of the water-tubes. If the tubes are nearly horizontal the circulation is naturally slow, and to prevent over-heating the diameter of tube and water space must be large ; but if the tubes are upright, the diameter of tube and water

space may be small; and herein lies the essential difference between so-called *large-tube* and *small-tube* water-tube boilers. Now consider a space 1 square yard in area and 8 feet long, and it is evident that a much greater number of small tubes than large tubes can be fitted into it, allowing the same distance between any two adjacent tubes; in other words, a much larger boiler heating surface can be fitted into an equal space and volume; and in destroyers, and recently in the largest vessels, this has been done with a saving in weight as well as volumetric space. In water-tube boilers the brickwork and casings are of considerable importance, and on their continued efficiency depends the efficiency of the boiler, other things being equal. In a *large-tube* boiler with nearly horizontal tubes, such as the Belleville, Babcock, Niclausse, and Dürr boilers, the brickwork and casings around the furnaces are exposed to a temperature of 2500 to 3000 degrees F. on four sides; in the upright tube type, such as the White-Forster and Yarrow, this high temperature only touches the front and back. As the various boilers are described in this and other chapters, the differences become apparent; and, without prejudice, each boiler should be rated on its own merits.

The following table (*Engineering* of August 2, 1907) shows the effect of the rise of boiler pressure on the coal consumption, power and speed, etc., of a few typical Cunard Atlantic Liners:

	1840. *Britannia.*	1856. *Persia.*	1879. *Gallia.*	1884. *Umbria.*	1893 *Campania.*	1907. *Mauretania.*
Steam pressure (lb.)	9	33	75	110	165	200
Coal necessary to steam to New York (tons)	570	1400	836	1900	2900	5000[1]
Coal per I.H.P. per hour (lb.)	5·1	3·8	1·9	1·9	1·6	1·45[1]
I.H.P. per ton of coal per hour	440	590	1180	1180	1400	1545[1]
Indicated horse-power	710	3600	5000	14,500	30,000	68,000
Speed (knots)	8·5	13·1	15·5	19	22	25[1]
Displacement (tons)	...	...	...	10,500	18,000	38,000

[1] Estimated—no actual details published.

Return-tube Cylindrical Boiler.—A circular shell pressed outwards by an internal pressure uniformly distributed over its inner surface, as in a boiler, tends to maintain its shape. This principle is embodied in the cylindrical boiler; but the flat front and back tend to bulge out, like a sail under the pressure of the wind, and to prevent this happening stays are fitted from front to back. These, as shown

in Fig. 26, however, all run parallel to each other, and access can be obtained to the interior of the boiler if they are arranged at convenient distances. The circular shell has been used for pressures up to 300 lb. per square inch, but, as will be noticed later, the thickness of the plates then becomes somewhat too great for convenient manipulation in manufacture. The weight increases considerably with the pressure, rendering their adoption for general use a matter of some doubt for pressures above 220 lb., or in special cases where weight is of great importance.

In Fig. 26 a cylindrical boiler, used for high pressures, is shown with a front view in part section and a side view wholly in section.

There are three furnaces (in some cases four are fitted, and sometimes only one or two) : *K* is a fire door, *F* is the ashpit, *AB* is a firegrate on which the fire is lighted, *C* is the combustion chamber, and *DD* the tubes leading to the uptake to the funnel. The boiler is filled with water to a few inches, generally 7 to 10, above the top of the combustion chamber, *i.e.* the highest part of the heating surface, and the space above the water-level is called the steam-room, or *steam space*. Fires are lighted on the firegrates inside the furnaces, and after they are sufficiently burnt through the fire doors are closed. The air necessary for combustion passes through the space *F* below the firegrate ; a swing door, opening inwards at the bottom, is fitted at *F* to regulate the draught. The burning gases pass from the fire into the combustion chamber, and then through the tubes, internally, to the smoke box, and thence to the uptake and the funnel.

The furnaces, combustion chamber, tube plates, and tubes, which are called the *heating surfaces*, are heated by the flames and hot gases, and these in turn communicate their heat to the water, which is in contact with them all ; thus steam, generated from the water, eventually fills the space above the water-level, and is compressed in the steam space until the necessary pressure is reached, when it is conducted away as required.

Circulation and Water Spaces.—Hot water is lighter than cold, and naturally tends to rise ; in a cylindrical boiler, therefore, the circulation is upwards above the furnaces and other heating surfaces of the combustion chamber and of the tubes. The movement of the mixed water and steam induces a downward current at the colder parts, such as the circular shell and the front of the boiler. In the modern type of cylindrical boiler, shown in Fig. 26, it will be noticed that—

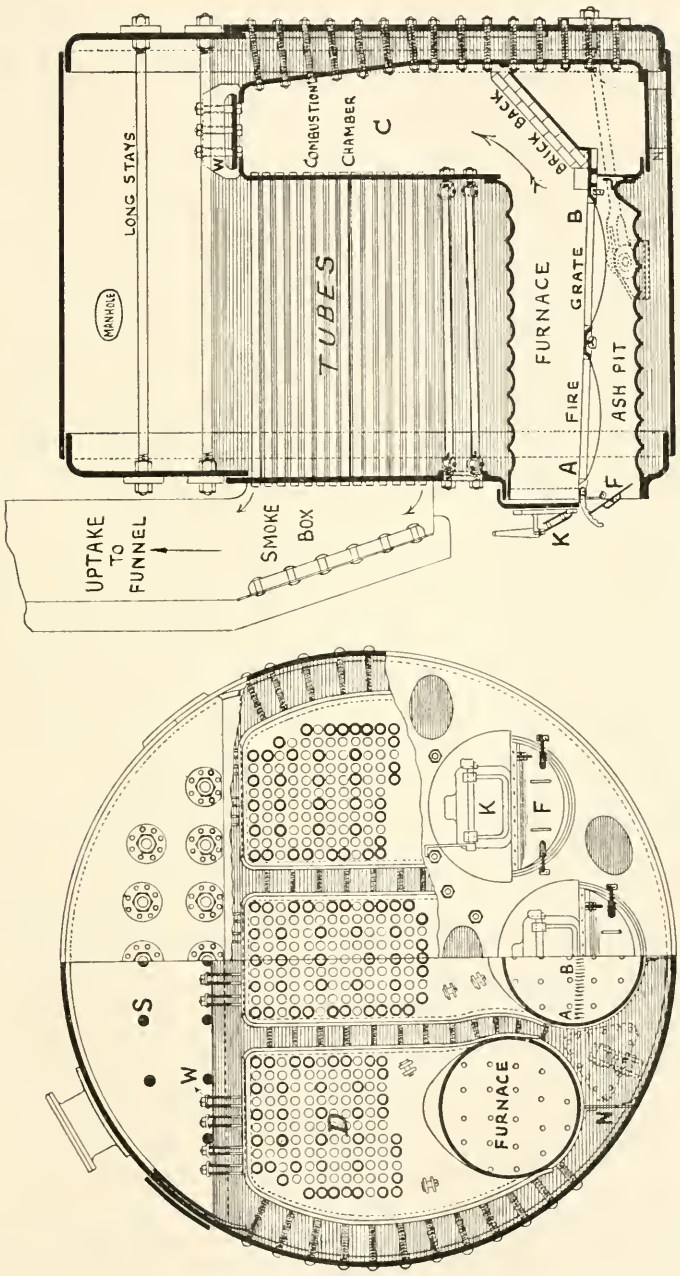


FIG. 26.—Cylindrical Return-tube Boiler. Coal fired.

(*a*) The water spaces widen at the top, where it is possible to fit them so.

(*b*) There is a large space above the tops of the furnaces clear of tubes.

(*c*) There is a good space between the nests of tubes over each furnace.

(*d*) The tubes themselves are spaced a fair distance apart, generally 1 inch vertically and from 1 to $1\frac{1}{4}$ inches horizontally.

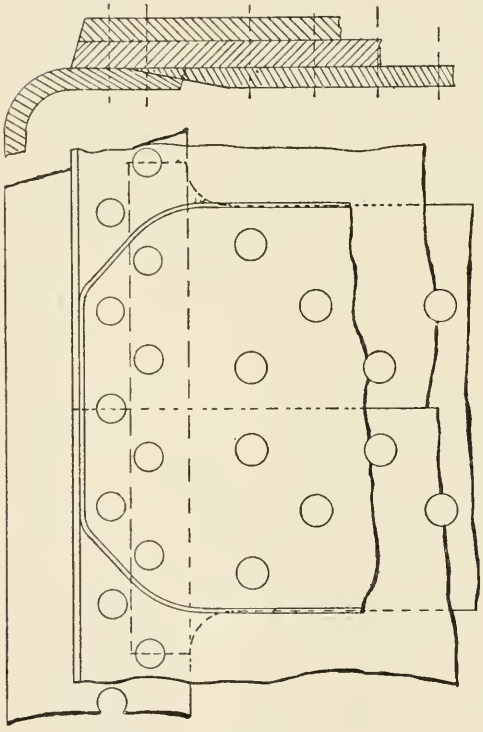


Fig. 27.—Boiler Shell Riveting.

(*e*) The furnaces are also arranged to be as great a distance apart as is consistent with their fitting in the smallest possible shell.

Many of the early boilers used under forced draught failed because the water spaces were omitted between some of the combustion-chambers, thus restricting the circulation to such an extent that the steam could not readily escape upwards, and so caused local overheating of the tube ends and plates. In all later types there is a water-space around each combustion-chamber, and each furnace has a separate combustion-chamber. Attempts were made to separate the common combustion-chamber by brickwork; but as this does not touch the real evil, such brickwork was an entire failure and tended rather to increase than decrease the leakage. (See also Ferrules, Double-ended Boiler.)

Construction of Shell.—As seen in Fig. 26, the modern boiler is arranged with the circular plate in *one* width, with the longitudinal joints clear of the screwed stays and the boiler seating. The longi-

tudinal seams are treble riveted, as shown in Fig. 27, with double butt straps. Double riveting is used for the junctions of the shell and end plates, and for the junction of the separate plates of the ends. The joints of furnaces and combustion chambers are generally single riveted; this forms a good tight joint, which is nearly all that is necessary, as there is little tendency to sheer at these points.

The strength at a riveted joint is less than that of the plate, because the holes in the latter, through which the rivets are passed, reduce the sectional area of the plate where the greatest stress is taken, which is along the line of rivets farthest removed from the edge of the joint, in proportion to the amount so removed. The strength of any structure is that of its weakest part, and consequently the strength of the boiler shell is governed by that of the joint employed. For the best system of riveting, the strength or efficiency of the joint is about 75 per cent

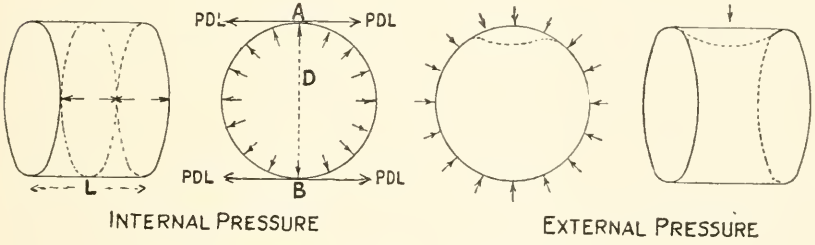


FIG. 28.—Effect of Internal and External Pressure.

of that of the plates so joined together if there are an equal number of rivets in each row, or 86 per cent if alternate rivets in the outer row are omitted, as shown in the Figure.

Shell Plates. — The shell of the boiler is subject to internal pressure, and if the circular shell be divided into two imaginary parts at any diameter, *AB*, the pressure tending to cause rupture at the extremities is that of a total pressure exerted at right angles to this diameter, as shown in Fig. 28.

Thus if P = the working pressure or load, in pounds per square inch on the safety valves;
D = the internal diameter of the shell, in inches;
L = the internal length of the shell, in inches;
t = the thickness of the shell, in inches; and
T = the tensile stress allowed on the plate, per square inch of sectional area in pounds;

then the total pressure tending to rupture the circular shell at the two extremities of any diameter AB—

$$= P \times D \times L ;$$

and this is equal to the resistance opposed to it, which—

$$= 2 \times t \times L \times T,$$

or

$$P \times D \times L = 2 \times t \times L \times T ;$$

from which—

$$t = \frac{P \times D}{2 \times T}.$$

The strength of the boiler shell to resist rupture longitudinally is greater than circumferentially. Adopting the previous notation, then the total pressure on one end of the boiler—

$$= P \times \frac{\pi \cdot D^2}{4} ;$$

and this pressure is distributed over the circular length of the plate, so that the resistance to rupture is—

$$= t \times \pi \cdot D \times T,$$

or

$$P \times \frac{\pi \cdot D^2}{4} = t \times \pi \cdot D \times T ;$$

from which

$$t = \frac{P \times D}{4 \times T},$$

which is only half the thickness required to take the circumferential stress ; the joints connecting the circular shell with the end plates are therefore generally quite strong enough when only doubly riveted.

Under Admiralty rules the thickness of the shell plating of a boiler about 13 feet diameter and for a working pressure of 210 lb. per square inch is $1\frac{1}{4}$ inches. By applying the above formulæ, it will be found that any increase in working pressure or diameter produces a large increase in thickness of plate, and consequently of weight. Until recently the manufacture of plating suitable for boiler shells was limited to a thickness of about $1\frac{1}{4}$ inches, but they are now made up to a thickness of about $1\frac{3}{4}$ inches.

Boilers constructed under the rules of the Board of Trade, Lloyd's, British Corporation, and other authorities, are slightly thicker than under Admiralty rules for the same working pressure ; the principle on which the thickness of plating is calculated is, however, similar. Boilers have been constructed for working pressures up to 300 lb. per

square inch. As an example of high pressures, the *Inchmarlo* has cylindrical boilers of 13 feet diameter, constructed with shell plates $1\frac{19}{32}$ inches thick for a working pressure of 267 lb. per square inch; the thickness would be about $\frac{1}{32}$-inch less under Admiralty rules.

Usually the thickness of the furnace plating has determined the size and working pressure rather than the thickness of the shell plates, which can generally be adapted to any pressure which can be used for the furnaces.

Furnaces.—The furnace is subjected to *external* pressure, and to withstand this pressure it is made circular. If a furnace is absolutely perfect in form, no distortion from a uniformly distributed external pressure is likely to take place, but such perfection is almost impossible to attain in practice, and consequently a large margin, or factor of safety of about six times the designed pressure, is adopted. Consider a furnace not quite circular or cylindrical, with its horizontal diameter greater than its vertical diameter; then the resolved vertical force or pressure tends to produce collapse, because it is not perfectly balanced by the resolved horizontal force or pressure. On the other hand, if the pressure were *internal* it would tend to produce circularity as in a boiler shell plate. In practice, if the distortion or difference in diameter of a furnace exceeds twice the thickness of the plate, it is considered that repairs are necessary, and usually the furnace is heated locally as necessary when pushing or setting back the furnace to its original circular form.

With low pressures the plain cylindrical furnace is strong enough to resist collapse when made of moderate thickness. An increase in thickness causes a loss in the rate of conduction of heat through the plate, but only in extreme cases is it likely to lead to overheating of the plating itself. In an ordinary boiler, from 50 to 60 per cent of the total heat absorbed by the boiler water is conducted through the furnace plates.

Various methods were employed to strengthen the furnace against collapse under pressure, and these led to the introduction of *corrugated* furnaces in 1887 by the Leeds Forge Company, of Yorkshire iron, but in 1883 they introduced furnaces made of Siemens-Martin acid open-hearth steel, of which the tensile strength has been gradually improved up to 26 to 30 tons per square inch, which is commonly used. In some few cases a tensile of 28 to 32 tons is specified and supplied.

The **Fox Furnace** is made with a number of uniform corrugations, each of which is of the same pitch and height, with a uniform thickness

throughout its entire length. The pitch of the corrugations is 6 inches and the depth is 2 inches, or a difference of 4 inches between the greatest internal and external diameters. The practical objections to the Fox furnace are that the scale accumulates at the bottom of the corrugations on the water side at the top of the furnace ; the opposite side of the plate at these points is exposed to the greatest heat of the fire ; and there is, in consequence of the difficulty in removing the scale, a great tendency to overheating with a reduced efficiency ; the furnace also tends to elongate too much under pressure and expansion,

SECTION OF PURVES' PATENT FURNACE IN TESTING APPARATUS.

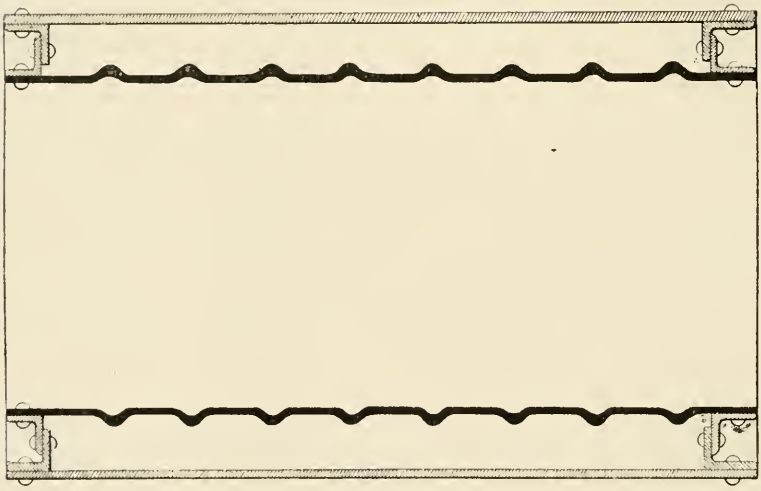


Fig. 29.—Purves Flue.

producing unnecessary stress on the boiler plating in connection with the furnace. The increased surface area is, however, in favour of all corrugated furnaces.

The **Purves Furnace** (Fig. 29) reduces the objections raised above, but the lack of uniform thickness is in itself objectionable from lack of uniform strength. The pitch of the corrugations is 9 inches and depth is 2 inches, and the thickness at the apex of the corrugations is considerably greater than the remainder of the furnace plating.

In the **Morison Furnace** the corrugations are pitched at 8 inches (203 mm.) and the depth is 2 inches (54 mm.) ; the plate is of uniform thickness throughout. The tendency to elongate under expansion

and pressure is not so great as in the Fox type. Fig. 30 shows this furnace arranged so that it can be removed without disturbing any other part of the boiler structure; but this construction has for very high pressures an objection in itself—it entails a very large opening in the front plate so that the corrugations can be passed through it. The flanging at the furnace front, generally outward in modern boilers, must be of such curvature that it can be made without appreciable loss of strength. Removable furnaces entail a boiler of somewhat larger diameter for the same width of firegrate, and there is consequently an augmented thickness of shell and weight. Some boilers are not arranged on the plan of removable furnaces, as the question

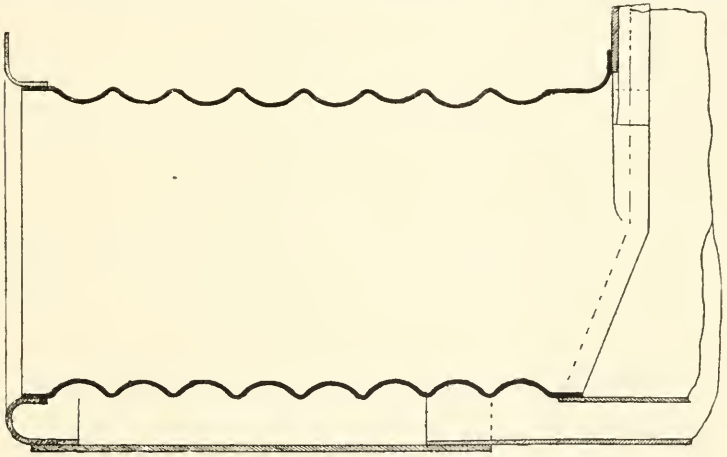


FIG. 30.—Morison Suspension Furnace.

of removal is not of great importance with the greatly improved methods of manufacture and material.

Deighton-Morison Furnace.—This is a combination of alternate small and large corrugations, as shown in Fig. 31. The pitch of the latter is generally 8 inches, and the difference between the greatest and least diameters is $4\frac{1}{4}$ inches. The furnace is lap welded and of uniform thickness throughout. The longitudinal elasticity is about one-third that of the Fox type, but it is sufficient for the necessary following of the expansion of the tubes. There is about 18 per cent more heating surface than in a plain cylindrical flue.

Brown's Cambered Furnace.—As fitted in some ships, the corrugations are pitched at about 9 inches apart, and the depth is about $1\frac{5}{8}$; the plating is about 45 per cent thicker at the ridges (as

in the Purves type), and the curve from ridge to ridge is similar in form to that of the Morison type, but in the "improved" type shown in Fig. 32 the thickness is uniform.

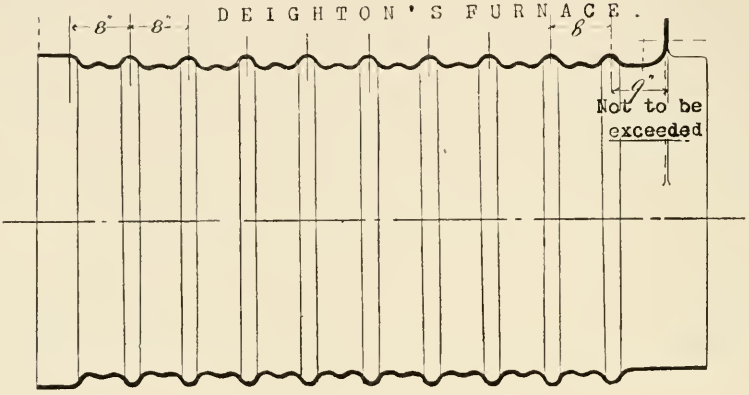


Fig. 31.—Deighton-Morison Furnace.

For nearly all cylindrical boiler plating, it is the common practice to fit plates about $\frac{1}{16}$-inch thicker than the specified minimum thickness. The extra thickness allows a fair margin for any inequality in

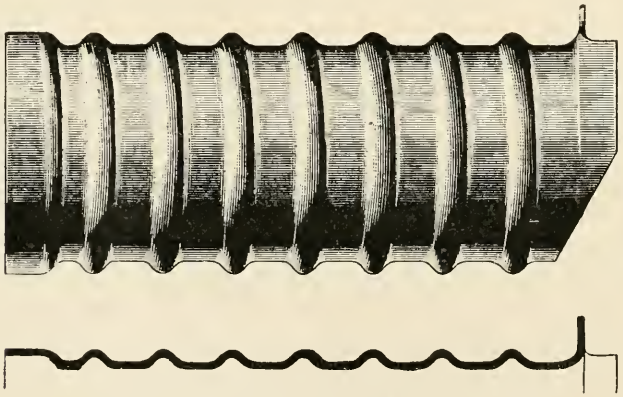


Fig. 32.—Brown's Cambered Furnace.

manufacture and for wear. For the same reason, stays are usually larger in diameter than the minimum specified.

Suspension Bulb Furnace.—Fig. 33 shows one of these furnaces

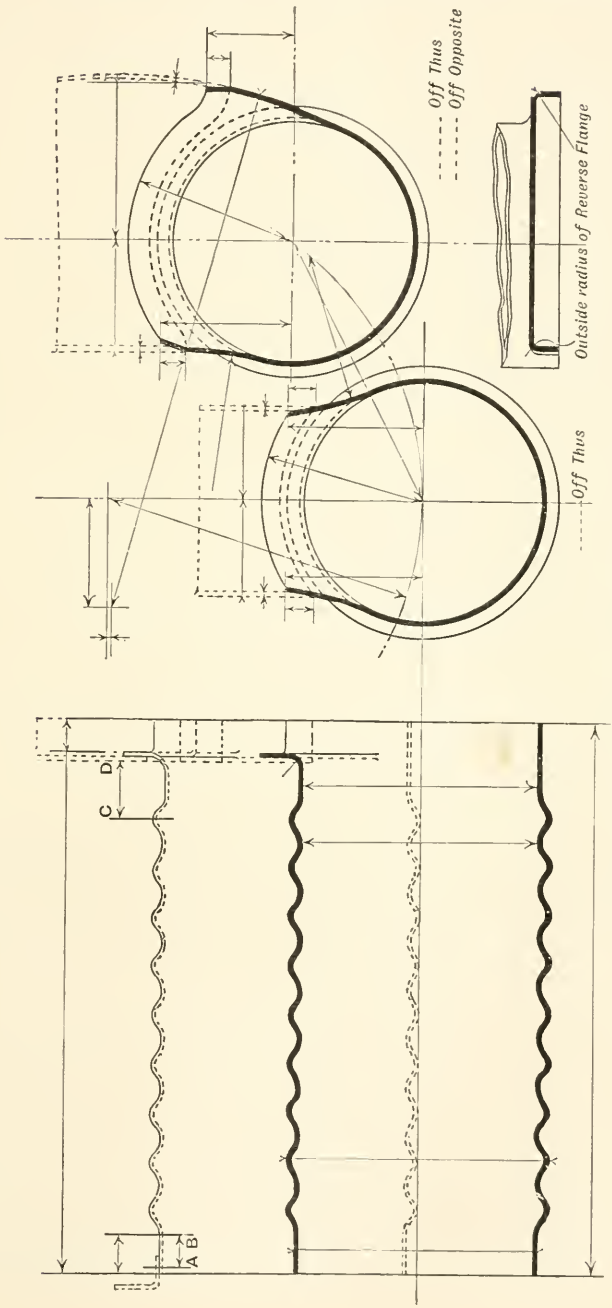


FIG. 33.—Data for Three-furnace Boiler (Leeds Forge Co.).

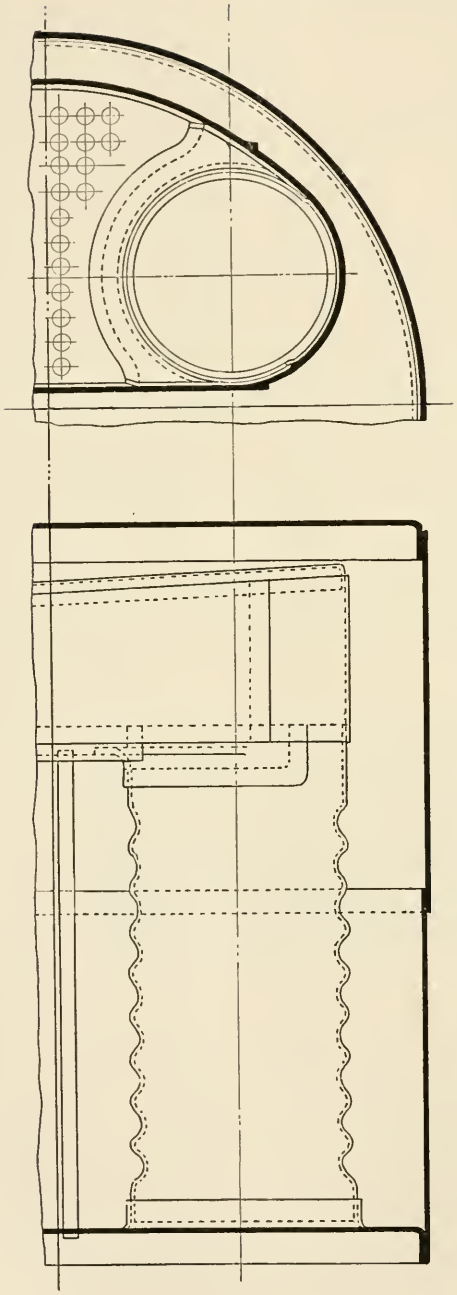


Fig. 34.—Interchangeable Furnace "O" Type (Leeds Forge Co.).

in section with an example of particulars for drawing a three-furnace boiler for manufacturing purposes. Fig. 34 shows another furnace of the same section arranged as Interchangeable "O" type (Leeds Forge Company).

These furnaces are machine made by the Leeds Forge Company, with a pitch of 8 inches (203 mm.), and a depth of corrugation of $2\frac{1}{2}$ inches (57 mm.). A furnace of this type, $46\frac{1}{2}$ inches in diameter (internal) and $\frac{9}{16}$-inch thick, successfully withstood a pressure of 1105 lb. per square inch before giving way, a result not yet approached by any other type of furnace in use.

Machine-made furnaces of the Suspension Bulb type, provided they are practically true circles, that the pitch of the bulbs does not exceed 8 inches, that the depth from the top of the bulbs to the least outside diameter is not less than $2\frac{1}{4}$ inches, that the plates are not less than $\frac{5}{16}$-inch thick, and the curves between the bulbs are fairly uniform in thickness, may be allowed under

British Admiralty and Board of Trade Rules. The working pressure is found by the following formula :—

$$\text{Working pressure (lb. per sq. inch)} = \frac{15{,}000 \times T}{D}$$

where T = Thickness in inches ;
D = *Least outside* diameter in inches.
Tensile of material = 26 to 30 tons per square inch.

For Fox, Morison, Deighton and other types the coefficient would be 14,000 instead of 15,000 in the above formula, which is based on a *factor of safety* of six times greater than the working pressure.

Stays.—All flat surfaces subjected to fluid pressure require staying to prevent distortion. The method used for a modern boiler under high pressure is shown in Fig. 26, which is in accord with the latest

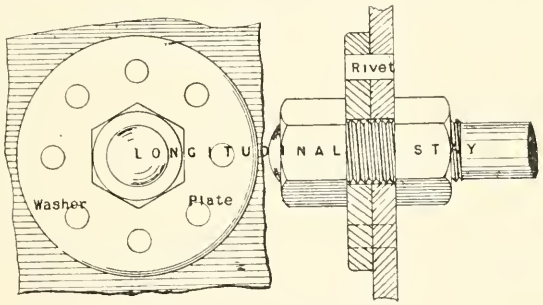


FIG. 35.—Longitudinal Stays.

developments, and the reasons for departure from the older types are detailed.

The long or *longitudinal stays* are fitted from front to back above the combustion-chambers (*S*, in Fig. 26), and an enlarged view is given of one end of a stay in Fig. 35. These stays are about $2\frac{1}{4}$ to $2\frac{1}{2}$ inches in diameter, and pitched at a distance of about 14 to 16 inches from each other at the corners of a square. They are solid from end to end, and not allowed to be welded ; at each end a *plus* thread is formed, and nuts are fitted (as shown) on both sides of the plate, with a thick riveted washer on the outer side to distribute the tension exerted by the stay over a sufficient surface to prevent local distortion. These washers are made of a diameter about two-thirds the pitch of the stays, and they are secured by rivets to the end plates ; the thread on the screw is standard Whitworth, about 4 to the inch.

Above the furnaces and below the tubes there is a large unsupported

space, so that staying is required. These stays are shown in detail for the end fitting in Fig. 36. They are generally made *portable*, for convenience of fitting and for removal when cleaning the furnace and other heating surfaces. The joint of the stay is therefore made with a pin and split pin, as shown ; and to allow for a certain estimated expansion, an elongated hole is made at the joint near the combustion-chamber end of the stay. The distance allowed for expansion is about $\frac{3}{32}$-inch for a length of furnace and stay of about $6\frac{1}{2}$ feet. The front tube plate between the nests of tubes has a certain unsupported area, because at the combustion-chamber end a water-space is formed between the several combustion-chambers. This space on the front plate is supported by *internal dog-stays*, as shown in Fig. 26.

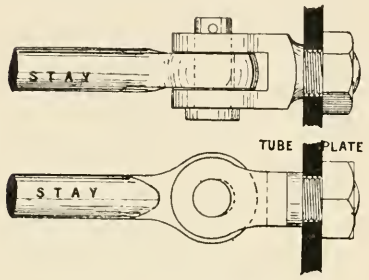


Fig. 36.—Portable Stays.

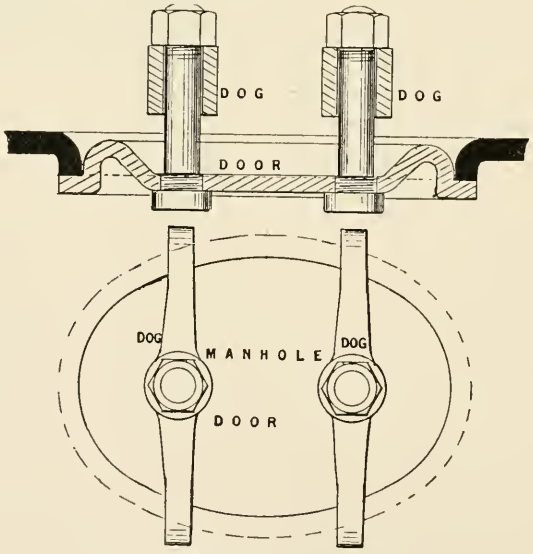


Fig. 37.—Manhole Door and Dog-stay.

The region of the front manhole doors, near the furnaces, is stiffened by flanging the front plate internally at these places, and the flanges form elliptical rings, on which the oval doors are jointed internally,

as shown in Fig. 37. The parts of the back shell plate near the bottoms of the combustion-chambers and opposite to these manholes are supported, as shown in Fig. 26, by portable stays, jointed to a T-piece riveted to the shell. No allowance is necessary for expansion at these points, and the connections are rigid.

Combustion-Chamber and Stays.—The tops of the combustion-chambers are flat, and need staying. *Vertical stays* were sometimes fitted to support the top of the combustion-chamber from the outer shell; a forked pin-joint was fitted to each end of each stay, which was connected with an angle steel riveted to the shell, and with a screwed and nutted stay in the top plate of the combustion-chamber. An elongated hole was generally fitted to the top pin-joint to allow for difference of expansion. When steam is up, the natural expansion

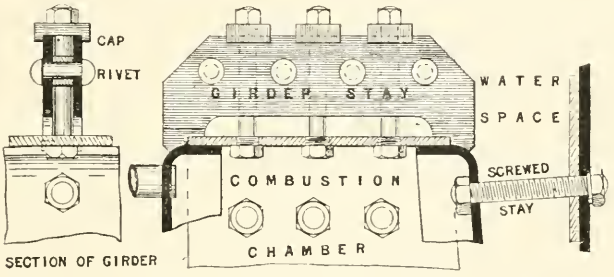


FIG. 38.—Bridge and Screwed Strays.

of the shell tends to raise it at the top, but the great pressure on the top of the combustion-chamber thus tended to pull the shell down and to distort it. The vertical stays are not supposed to come into play until the natural expansion has taken place, but the exact allowance at the pin-joint connection is always somewhat problematical. In all recent boilers of this type, *bridge or girder stays*, which do not interfere with the natural expansion, are fitted, as shown in Figs. 26 and 38.

The *bridge or dog stays* fitted on top of the combustion-chamber are sometimes made by riveting two plates together with a distance piece or ferrule between them. The bridge is supported by the front and back plates of the combustion-chamber as much as possible. Caps are fitted on top of each bridge (Fig. 38), and stays are screwed through the top combustion-chamber plate, and nutted above the caps. The stays are made with a *plus* thread under the heads, which make steam-tight joints with the plate, and the ends of the stays are riveted

over above the nuts, to prevent the latter from slacking back; this security is also used for other nutted and screwed stays. The size and pitch of stays are in all cases calculated for the stress they have to bear.

The sides and back of each combustion-chamber form large flat surfaces, and are stayed by means of *screwed stays* nutted on each end. The stays are screwed through both the plates they secure by a continuous thread (Fig. 38), and, as far as possible, it is arranged that the stay is perpendicular to the plate exposed to the heat of the flue gases; at the other end, the plate is either sufficiently recessed or a washer fitted to form a fair surface for the face of the nut. All nuts exposed to the action of the fire, steam, or hot water are generally made of steel; but all external nuts, such as those on the ends of the long stays and on bolts of manhole doors, are made of iron.

The parts of the back shell plate opposite to the water spaces, formed by the separation of the combustion-chambers, are strengthened by means of doubling plates, and stronger stays are fitted near these parts, connecting the back plate with the combustion-chambers.

The bottoms of the combustion-chambers are made semicircular in shape, and of rather thicker plate, so that stays are unnecessary; but an angle-steel frame N, connected with the boiler shell only, supports the weight of each combustion-chamber (Fig. 26).

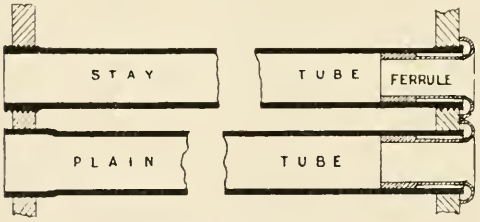


Fig. 39.—Plain and Stay Tubes—ferruled.

Tubes.—For forced-draught working the external diameter of the tubes is generally $2\frac{1}{2}$ inches, but if intended for natural-draught only they are sometimes larger. The thickness of the plain tubes is about 0·16 inch, and the ends at the smoke box are swelled out to $\frac{1}{16}$-inch greater diameter for facility of fitting and removal (Fig. 39). The stay tubes are of $\frac{1}{4}$-inch thickness, and screwed into the tube plates with a continuous thread; the front end is swelled out as with the plain tubes, and the thread is a *plus* one, so that the thickness at the bottom of the thread shall not be less than that of the tube in its plain part. The failure of some of the earlier boilers was due to the part of the tube on which the thread was cut being a separate piece welded on to the tube ends; the value as a stay was therefore very

small, especially in some cases where the weld was very imperfect. The thread used for stay tubes is a fine pitched one, about 10 threads to one inch. After the tube is screwed into its place it is expanded by a tube roller in the usual way, and caulking (which would disturb its proper fitting) is not allowed in good practice. The tube ends of both plain and stay tubes should not project more than a small necessary amount, say $\frac{1}{8}$-inch, beyond the tube plate at the combustion-chamber end ; this allows the cap ferrule to be fitted, as shown in Fig. 39.

In mercantile practice, *Serve tubes*, as shown in Fig. 40, are sometimes used for both plain and stay tubes. The object of their use is to increase the strength of the tube and at the same time to increase the heating surface. There are a number of ribs which extend for nearly the whole length of the tube, but which do not interfere with the usual rolling and expanding of the ends into the holes in the tube plates. The advantage gained is that for short periods a greater output of steam can be obtained from a smaller number of Serve tubes than from a larger number of ordinary tubes of the same length and diameter, but this advantage is entirely lost after the boiler has been under steam for a moderate number of hours and the tubes become choked with soot and small scoriæ. They cannot be easily swept or cleared when under steam, and for naval purposes have seldom been adopted. A more efficient method of increasing the output of the boiler is now commonly adopted in British practice, and is explained under the heading of Retarders.

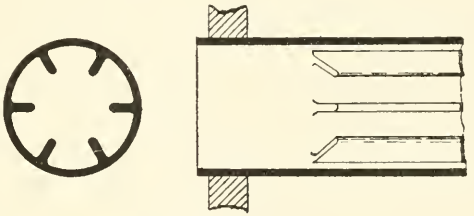


FIG. 40.—Serve Tube.

In a few water-tube boilers in foreign navies of the Belleville type, Serve tubes are used next the fire.

Ferrules.—The earlier boilers which gave trouble at the tube ends from leakage, under forced draught particularly, had a ferrule fitted in the combustion-chamber end of each tube ; the ferrule was simply a hollow cylindrical wedge, the outer end of which was about level with the end of the tube when driven home ; it was subsequently rolled and expanded into the tube at the tube plate. These ferrules served a very useful purpose, and in many cases kept the tubes tight when they might have failed without their assistance.

A large portion of the heat originally transferred from the tube

to the tube plate was still transferred through the medium of the ferrule, and still resulted in the accumulation of steam at the surface of the plate in such quantity that overheating took place, culminating in leakage at the tube end ; it mitigated, but it did not entirely cure the evil of tube leakage.

The Admiralty *cap ferrule* was then introduced, and practically stopped all tube-end leakage within fairly high limits. The ferrule shown in Fig. 39 is made to cover the end of the tube, and to form a nearly non-conducting air-tight space with the tube plate. The heat imparted to the ferrule is conveyed along it to the part where it is in contact with the tube, about $\frac{3}{4}$-inch distant from the inner face of the tube plate ; this decreases the ebullition near the tube plate, and the always limited circulation of steam and water more nearly suffices to prevent overheating. The ferrule is simply tapped into the tube end with the wooden mallet, and should not tend to distort the tube in any way.

Retarders.—The retarder consists of a piece of flat plate of the same width as the internal diameter of the tube, twisted into the form of a long spiral helix, about $2\frac{1}{4}$ turns in a length of 6 feet 6 inches ; it is pushed, not forced, into the tube until its inner end is near the end of the cap ferrule, and its outer end then projects just clear of the smoke-box end of the tube. The heated gases from the fire have now to travel around this helix formation, thus bringing the whole of the tube surface into play as heating surface.

The gases take from 2 to $2\frac{1}{2}$ times as long to pass through the tubes, and thus they are given more time to diffuse their heat. The efficiency of the boiler is increased by 2 to 8 per cent by the proper use of retarders. When sweeping tubes, the retarders are removed, and this generally assists materially in keeping the inner surfaces of the tubes clean and free from accumulation of soot and small scoriæ. For convenience of removal, each retarder is provided with a hole in its outer end.

Double-ended Return-tube Boiler.—When two boilers (Fig. 26) are placed back to back without the back plates and so form one structure, it is called a double-ended boiler, a type of which is shown in Fig. 41. This boiler is fired from each end, and the number of furnaces is sometimes as many as eight ; but generally six are fitted, three at each end.[1] In all recent boilers a separate combustion-

[1] The 25-knot Cunarder *Mauretania* is fitted with 23 double-ended boilers and 2 single-ended boilers, of the 4-furnace type, making 192 furnaces in all.

chamber is fitted to each furnace, for the reasons already given; if this is done, there is no objection to the double-ended boiler as compared to the single-ended variety, and there is some saving of weight and space by its adoption. There appears, however, to be some bias in favour of the single-ended return-tube boiler, which is probably due to its more extended use with more reliable results; double-ended boilers have recently been fitted in several large passenger vessels.

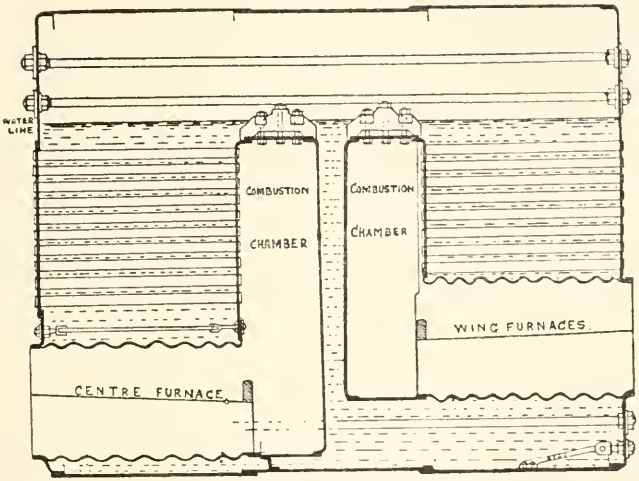


FIG. 41.—Double-ended Boiler, with Single Combustion-chamber.

Direct-tube Boiler.—Where the height was insufficient to fit the return-tube boiler, the tubes were arranged in line with the furnaces instead of immediately above them; this kept the outer diameter of the boiler much smaller, and saved the height at the expense of the ground space. They were fitted in many of the earlier ships of the small cruiser type and in gunboats, and generally they were successful for this purpose; thus H.M.S. *Porpoise*, fitted with this type of boiler in 1885, was continuously employed for twenty years. The ground space and extra weight do not compete with modern types, and in all the smaller vessels for many years past upright water-tube boilers have been fitted, with increasing success and efficiency.

Locomotive Boiler.—This type of boiler has been succeeded by water-tube boilers. It is a direct-tube boiler, and its success in locomotive practice entitles it to great respect; but the conditions of marine working are so different, as pointed out later, that although

succeeding on shore it has been a distinct failure on board ships for high rates of forcing.

Efficiency of the Boiler.—The complete combustion of a given quantity of fuel will generate a certain quantity of heat. In practice the combustion is not quite complete, and a certain portion of the fuel passes away unconsumed, leaving only the heat generated by the remainder for possible transmission to the water in the boiler through the heating surfaces.

The *efficiency of combustion* is therefore the ratio or fraction of the heat available for transmission to that contained in the fuel.

Of the heat available for transmission, only a certain amount is actually transferred to the water in the boiler ; the remainder passes away, and is lost principally up the funnel.

The *efficiency of the heating surface* is the ratio of the actual heat transmitted to the water to that available for transmission.

The *efficiency of the boiler* is the product of these two efficiencies of combustion and of the heating surface ; or it may be given as the ratio of the heat actually transmitted to the water to that contained in the fuel.

$$\text{Efficiency of the boiler} = \text{efficiency of combustion} \times \text{efficiency of heating surface}$$

$$= \frac{\text{heat available for transmission}}{\text{heat energy of fuel}} \times \frac{\text{heat transmitted}}{\text{heat available for transmission}}$$

$$= \frac{\text{heat transmitted to water}}{\text{heat energy of fuel}} = 60 \text{ to } 78 \text{ per cent in good average practice.}$$

In some instances the steam pipes and fittings are included as part of the boiler, and then—

$$\text{Efficiency of the boiler} = \frac{\text{heat supplied to engine}}{\text{heat energy of fuel}},$$ which is obviously rather less than the true efficiency.

Weight of Cylindrical Boilers.—Until about 1890 there was a gradual decrease in weight of boiler per I.H.P., because the use of forced draught and better materials, and the increased economy obtained in the engine by using higher working pressure, counterbalanced the increase of weight necessary for the proper strength of the boiler and engine.

When the working pressure was increased from 150 or 160 lb. to 210 lb., the weight of cylindrical boilers increased with the pressure.

With these boilers the weight increases nearly in proportion to that of the shell plating, or—

$$\text{Weight} \propto t \propto \frac{P \cdot D}{2 \cdot T};$$

and for boilers of the same diameter D and strength T of material—

$$t \propto P.$$

Thus, if the pressure be increased from 200 to 300 lb. per square inch, the increase in weight of shell

$$= \frac{300 - 200}{200} = \frac{100}{200} = 50 \text{ per cent};$$

but the fittings, casings, and other parts are the same for both installations, and their weight is about 60 per cent of the total weight included in that of the boiler rooms; so that the actual increase is—

$$\frac{100 + 50 + 60 - 100 - 60}{100 + 60} = \frac{50}{160} = 31 \text{ per cent about.}$$

Although the size of engine is decreased with increased pressure there is a slight increase in weight, as greater strength is necessary for all parts through which the steam passes.

There are various ways in which boilers can be compared in relation to their weight:

(1) *Weight of water evaporated* per hour per ton weight of boiler.—If the rates of combustion be equal, the comparison is a very fair one; but proper means must be taken to ascertain the quality of the steam on leaving the boiler in regard to priming, or degree of moisture carried away in the steam.

(2) *Grate surface per ton* of weight.—There may be considerable difference in the length of grate; thus for the mercantile marine it may be only 5 feet or less, whereas in a warship it is sometimes 7 feet or more. A short grate is generally more efficient for continuous steaming.

(3) *Heating surface per ton* of weight.—The fairest comparison can be made on the heating surface provided per ton of total weight, including water at working height in the boiler. Some parts of the heating surface are more efficient than others, but for general purposes of comparison this does not materially affect the result. With a moderate rate of combustion the efficiency of the heating surface (and relatively, the boiler) is generally greater than with a very high rate.

(4) *I.H.P. per ton* of weight.—This is a fair comparison only when the rate of combustion and the weight of steam used per S.H.P. or I.H.P. in the engine are the same for each plant compared. The rate of combustion varies from about 15 lb. per square foot of grate in a tramp steamer to 220 lb. in an express locomotive, while the weight of steam used per S.H.P. varies from 10 lb. per hour in a few of the most efficient steamers to about 16 lb. in a destroyer. Recent cruisers and battleships with turbine engines use about 14 lb. weight of steam for *all purposes*, of which about 10 lb. are for the main engines, per S.H.P. per hour; new destroyers use about 11 lb. weight of steam for *all purposes*, of which about 10 lb. are for the main engines, per S.H.P. hour at full power, but the steam consumption increases per S.H.P. at powers below about one-half the full power, although naturally the total consumption becomes considerably less at smaller powers until a limit of about 10 knots is reached, when the steam and fuel consumption per mile travelled begins to increase, principally caused by the heavy auxiliary consumption and by the main engines being too large for the economical development of very low powers.

CHAPTER VII

UPRIGHT WATER-TUBE BOILERS

Circulation — Above Water Delivery. — The circulation caused by convection in boilers with below water delivery has already been explained in Chapter IV. The Belleville generator, the Niclausse boiler, the early Thornycroft boilers, and to some extent the Babcock

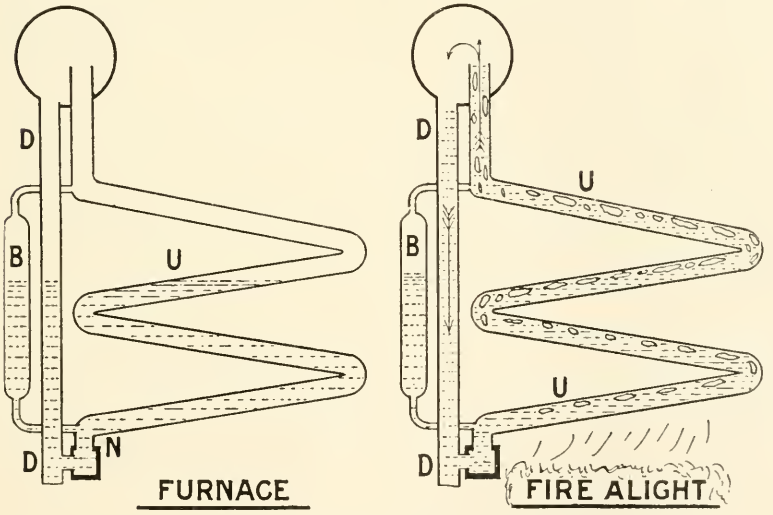


FIG. 42.—Circulation: Belleville Generator.

and other boilers, are so arranged that the mixed steam and water coming from the generating tubes are discharged into the steam chest above the normal working water-level.

In the Figs. 42 and 43, the water-level, *when cold*, is shown in outline diagrams of the Belleville and Thornycroft (old) types. After the fire is lighted the water in the tubes *U* becomes gradually hotter and expands. This causes an *apparent* rise of water-level in *U* ; but

as the weight (and, therefore, the head of pressure) is not increased, the water-level in *D*, which is little exposed to heat, remains unchanged. Steam bubbles soon begin to form in the water in the tubes *U*, and push the apparent water-level in them higher and higher until an overflow into the steam chest takes place. Up to this point no change has taken place in the level of water in *D*, because no change of head of pressure has taken place in either tube. At the instant that a discharge from *U* into the steam space takes place, the balance of head is upset ; that in *U* is decreased, and that in *D* is increased by any water discharged into it from *U*. (Any steam which is formed accumulates in the upper part of the steam chest, and such pressure that it may exert is equally supported by the water in *D* and in *U*.) The greater head created in *D* causes a movement from *D* to *U*, and starts a circulation which is continuous in direction, although the discharge of steam and water from the tubes *U* into the steam chest is generally intermittent.

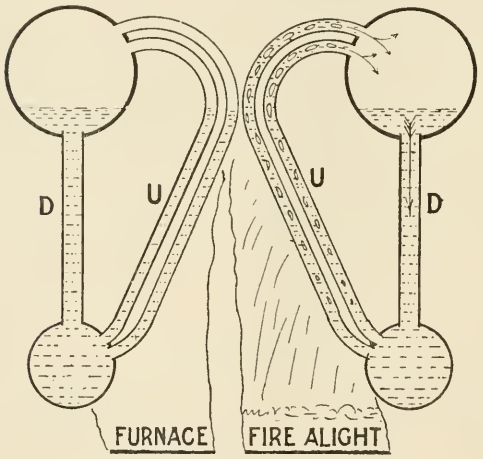


FIG. 43.—Circulation: Thornycroft (old type) Boiler.

In the Thornycroft boiler the level of the water in *D* remains practically constant, and is that of the working water-level. In the Belleville, however, it is probable that the level in *D* rises until it reaches the bottom of the steam chest ; with this change in level, there is an increase of difference of head.

In the Thornycroft boiler the circulation is free and unrestricted ; there is only one length of tube (from 10 to 15 feet long), and no acute changes in direction. The circulation is therefore rapid although the difference of head is small ; from model experiments, Sir John Thornycroft calculated that each particle of water passed through the steam chest 105 times before leaving it as steam.

In the Belleville boiler the frictional resistance to the circulation is great, owing to the nearly horizontal placing of the tubes—there are also about 70 to 80 feet of length of tube with about fourteen

to sixteen acute changes in direction,—and the circulation is restricted by the small passages N at the nipple connections. The circulation is therefore comparatively slow.

In all cases circulation, when once started in any one direction, tends to continue in the same direction; this is due to convection and change in density (as described above) producing difference of head; to momentum caused by this difference; to further increases of head and momentum caused by discharge of water from the generating tubes into the steam chest; and to the (kinetic) energy of motion produced by the momentum (force).

Downcomers.—Boilers built for the Navy were originally fitted with separate downcomer tubes, of fairly large diameter and remote from the fire and intense heat. They were generally fitted outside the casing enclosing the generator tubes. For a large boiler two downcomers of about $4\frac{1}{2}$ inches diameter were usually fitted to each water drum, either back or front, as convenient. Downcomers were considered necessary for marine purposes, when the ship rolled or pitched, but they have been safely dispensed with for boilers of the Yarrow, White-Forster, and other upright tube types.

If all the generating tubes are curved the downcomers may be straight, because expansion is thus allowed independently; but if the tubes are straight, as in some Yarrow boilers, the downcomers should be curved about $\frac{1}{4}$-inch in each foot of length.

Advantages of Water-tube Boilers.—Advantages claimed for water-tube boilers compared with the ordinary cylindrical marine type are briefly considered as below:—

1. *Strength.*—If sufficient weight is allowed, it is now possible to build a boiler of any type for any pressure up to about 360 lb. per square inch. If the weight is limited the strongest boiler is one of the water-tube variety, and with which the smallest tubes can be used with safety.

2. *Safety.*—Comparatively a very small weight of water is carried in a water-tube boiler at working height, and its sudden release, by fracture, is likely to be less disastrous than a larger quantity. The weight of water carried varies, according to type and size of tubes, from about one-fourth to one-third that of a cylindrical boiler of equal evaporation.

3. *Lightness.*—The smaller quantity of water carried at working height, and the absence of any heavy shell, reduce the weight of the boiler, power for power. There is a great difference in the weight of

water-tube boilers of various types, but generally the lightest boiler is that with tubes of the smallest diameter which can be safely employed.

4. *High Power.*—The power which can be obtained from a boiler depends on the area of heating surface which can be provided within a limited space and weight, and the greatest area of heating surface is obtained by using water-tube boilers with tubes of the smallest permissible diameter. A larger proportion of the ground space occupied by a water-tube boiler can be utilised as firegrate or combustion space, and, consequently, the rate of combustion need not be so high, power for power and space for space, as in a cylindrical boiler, and a higher percentage of the full power can be maintained for longer periods with a lower rate of combustion, because the fires do not require clinkering so frequently. One double-ended cylindrical boiler with eight furnaces may be designed to develop 3000 i.h.p. as compared with one upright-tube water-tube boiler developing upwards of 9000 s.h.p., occupying less space and of less weight.

5. *Facility of Repair.*—Permanent defects are frequently produced by sudden alterations of temperature in cylindrical boilers. The water-tube boiler is not usually affected in this way, as some provision is made for independent expansion of the various parts. (Frequent and undue haste in cooling is not conducive to the long life of any boiler; statements made in the opposite sense must be considered as comparative, as pointed out at the beginning of this section, and not absolute.) Fires can, therefore, be withdrawn and the steam and water blown out without any delay in cooling the plant, and any defects made good. A tube or element can be made good in a comparatively short time, and, in most of the boilers with upright tubes, a tube can be plugged in only a few minutes. The defects usually met with in cylindrical boilers are of a serious nature, and require considerable time to make good.

6. *Durability.*—As the generating tubes constitute the principal part of a water-tube boiler, its life practically depends on the life of the tubes. So far as present experience goes, the life of the tubes in a water-tube boiler is greater than of those in a cylindrical boiler, when used with forced draught of the same intensity.

The life of the tubes depends on:

(*a*) The thickness allowed for wear, especially of the tubes in the rows next the fire.

(*b*) The inclination of the tubes and the rate of circulation through

them. An upright tube, with a comparatively rapid circulation, is less liable to overheating, and consequent reduction in material and strength, than a horizontal one. The deposit of scale is lessened by a quick circulation.

(*c*) The purity of the water used for feeding the boiler, and its freedom from grease and acidity.

(*d*) The class of material, and the treatment of those in charge.

The generating tubes in destroyers and small vessels, where weight is of primary importance, are thinner than in other and larger vessels, and consequently their life is shorter. Even in these small vessels the margin for wear and tear is considerable, and instances are known where the decrease in thickness has been difficult to determine after some years' service. For examination a few tubes are withdrawn and thoroughly cleaned and scraped inside and outside ; they are then compared by weight with new tubes of the same length and thickness. Further examination is made by splitting and cutting up a certain number of tubes into short lengths, and then carefully gauging them.

From the foregoing it is evident that the upright-tube boiler can be made more durable than the nearly horizontal type, and that it possesses some tangible advantages, such as saving in weight, economy, and in many instances simplicity. Further development in upright-tube boilers is very probable for both naval and mercantile purposes.

7. *Handiness.*—The furnaces in water-tube boilers are all of one convenient height for stoking. In a cylindrical boiler the heights differ by $1\frac{1}{2}$ to $2\frac{1}{2}$ feet. Steam can be raised quickly, and the time required for raising steam is only limited by that required for warming the engines. If the engines are already moving, more boilers can be quickly added to those already in use.

The rapidity with which steam can be raised depends on the quantity of water which must first be heated to produce steam, and in water-tube boilers this quantity is considerably less than in the water-tank type.

8. *Economy of Fuel.*—In time of war, ships frequently cruise at moderate speeds, and the higher speeds are only required at uncertain intervals, at short notice. High speed at short notice necessitates keeping all, or nearly all the fires in cylindrical boilers alight or banked, and a consequent steady diminishing of the supply of fuel. It is unnecessary to keep the fires banked in boilers in which steam can be raised quickly, and the fuel otherwise burnt is thus saved, and the

steaming distance and period which a ship can keep the sea, without fresh supplies of fuel, are thus greatly increased.

"In the war with Russia it was found that Japanese ships with cylindrical boilers consumed *five* times the coal burned by those with water-tube generators, owing to the fact that, having to be ready at two hours' notice, they had to keep fires going while the Belleville boilered ships were able to let fires out."—*Heresies of Sea Power*, by Fred. T. Jane, 1906.

Generally, and except in cases of extreme urgency, fires should *not* be banked, but allowed to burn down for from six to eight hours, and then just kept alight near the furnace door and ready for pushing back; the boiler should not be cooled suddenly or quickly. About 14 cwts. to 1 ton of coal is required for raising steam in each boiler, and when banked at one hour's notice about 2 cwts. of coal is required per hour, and about double this quantity at short notice. It is therefore more economical to let the fires burn out, if steam is not probably required for about twelve hours, and when about one hour's notice can be given. During this time the cleaning of the fires and furnaces, sweeping tubes, possibly minor repairs, and preparation for steaming by re-laying the fires, may be carried on so that the ship is ready for full power, as opposed to something less with dirty fires and heating surfaces.

With, say, one-half the boilers alight and engines moving slowly, and therefore warmed through, a ship should be able to obtain in a few minutes nearly four-fifths of her maximum speed; but to maintain this speed more boilers must be added to allow for cleaning fires, etc. It appears, then, that it is more economical, and conducive to obtaining full power, to keep only a few boilers under steam or banked at short notice, and the remainder in all respects clean and ready, but with fires not alight. It is generally more economical, when high speeds are required at very short notice, to keep the engines moving slowly than to move them from time to time; this is because it requires some time to warm the engines through, and because the safe conduct of the operation of adding fresh boilers, especially in times of urgency and emergency, should be controlled by one head. Celerity with safety cannot be attained by drill in matters of boiler or machinery control, but both are possible by a cool and experienced head. "Move with caution but move promptly" is, perhaps, the keynote of successful practice; but promptness without caution produces more accidents and makes more defects than any other cause.

White-Forster Boiler

Figs. 44 and 45.

Typical Features. — There are two water-chambers and one steam collector. All the tubes are drowned — that is, they deliver their contents below the working water-level. The radius of curvature

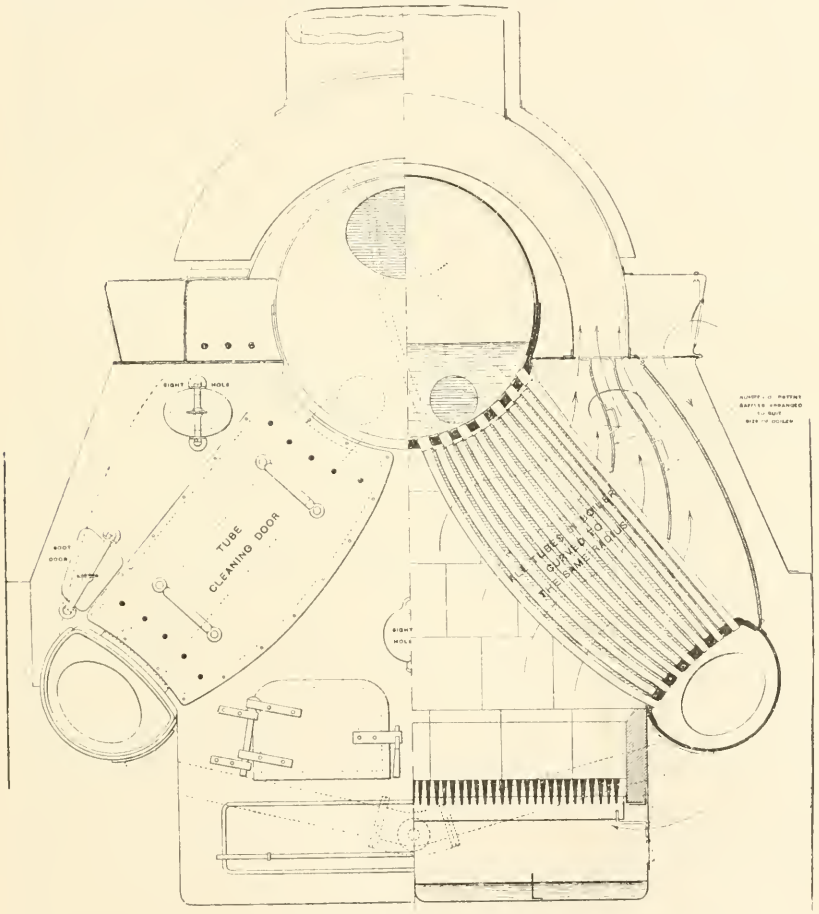


Fig. 44.—White-Forster Boiler.

of each tube is the same, and the curvature is only sufficient to determine the direction of movement when expanded by heat. The tubes are arranged like the staves of a barrel, and the continuation of the line of curvature of each tube, passing through the steam chest, allows

each tube to be withdrawn into and from the steam chest through a manhole. Each tube can therefore be withdrawn without disturbing any other tube. The tubes can be cleaned internally, as shown in the figure, by a tube brush whose rigid handle is of the proper curvature. This system of cleaning is probably the best yet devised.

Circulation.—Large downtake tubes are provided at the back end of the boiler, and the circulation is unrestricted. No separator plates are fitted, and the usual internal steam-pipe is fitted. A feed regulator is generally fitted to each boiler.

Path of the Furnace Gases.—The uptake is divided near the tubes by patent baffles, as shown in the figure, which have the effect

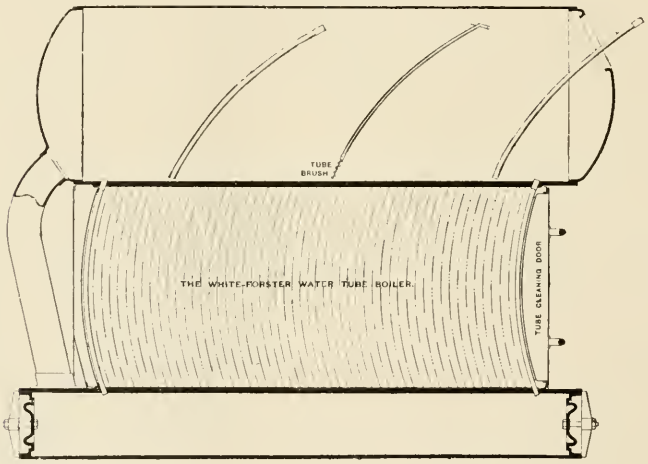


FIG. 45.—White-Forster Boiler.

of drawing gases equally over the whole of the tube surface, and at right angles to it. The number of baffles is arranged to suit the length of the tubes and conditions of working. Large doors are fitted at the stokehold end, so that the whole length of tubes is exposed for cleaning, and a brush can be passed from end to end of the boiler between the rows of tubes while the fires are alight.

Remarks.—The good features of this boiler are: the slight curvature of the tubes, the facility of withdrawal and renewal of the tubes, and the practicability of cleaning the tubes, both externally and internally. The boiler is sometimes constructed to be fired from opposite ends, and is then known as the double-ended type. In all recently constructed boilers the water drums are circular.

Yarrow Boiler

Figs. 46 and 48.

Typical Features.—The generating tubes are fitted between one upper steam chest and two lower water-chambers. The tubes are inclined at any convenient angle to suit the proposed height of boiler and width of firegrate, and generally the inclination lies between 45° and 60° to the horizontal. The tubes, when first fitted, are usually straight; but, after steaming, are generally found curved, to a more or less extent, by the action of the fire and difference of expansion. The continual change of form does not conduce to durability, and in boilers which are subjected to high rates of forcing it is now usual to make the two rows of tubes next the fire with sufficient curvature to allow for independent expansion.

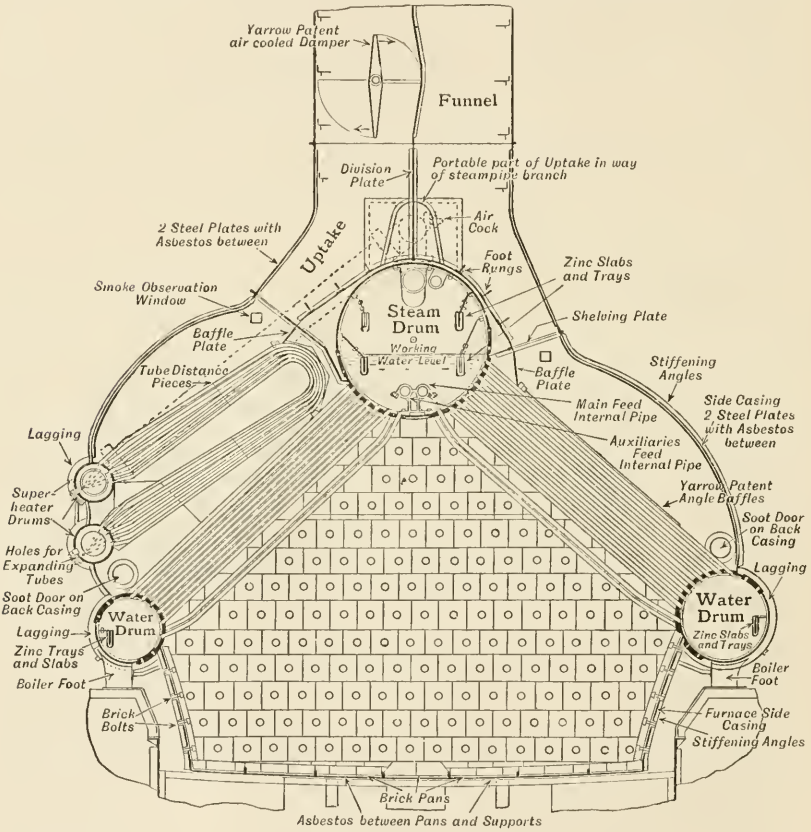
Circulation.—When downcomers are fitted, the course of circulation is similar to that of other boilers with downcomers. In boilers in which no down tubes are fitted the circulation is various, because at different rates of combustion, or conditions of rolling or pitching, a tube may be either an uptake or a downtake under the altered conditions. On shore, when the boiler is perfectly steady, the circulation, having once started up the tubes nearest the fire, and down some of those more remote from the fire, continues in the same direction, and its rate of movement accommodates itself to the necessary conditions.

In a few cases, instead of fitting large outside downcomers, some of the small tubes are made to act as downcomers by forming pockets with thin baffle plates inside the water-chambers. If the tubes in the wings, remote from the fire, are thus screened off, the supply may be temporarily stopped by the ship heeling; and if a few tubes at one end are used as downcomers, the allowance for expansion, between these comparatively cool tubes and the others acting as steam generators, is inadequate when all the tubes are straight.

Path of the Furnace Gases.—Fig. 46 shows the latest type of Yarrow boiler with superheater, which is fitted on one side only, and the number of rows of generator tubes is in consequence only twelve on that side compared with eighteen on the opposite side, without the superheater. About one foot from the top end of the generator tubes on each side a baffle is fitted to prevent a too easy exit for the furnace gases. On the superheater side this baffle is continued round the ends, of U shape, of the superheater tubes. On the other side, the baffle is continued downwards by Yarrow patent angle baffles for

the greater part of the length of the generator tubes, but leaving a clear passage of about 15 inches next the water drum.

Distance pieces, as shown, are fitted among the superheater tubes and these act as deflection baffles for distributing the gases evenly over the heating surfaces both of the generator and superheater tubes.



Transverse Section

FIG. 46.—Yarrow Water-tube Boiler, with Superheater.

The draught through the uptake on the superheater side can be checked by a Yarrow patent air-cooled damper, which if closed or partially closed would compel the greater part of the gases to discharge on the non-superheater side to the funnel.

Remarks.—The boiler shown in Fig. 46 is arranged for oil fuel firing, but could be adapted for coal if necessary, and otherwise the

general arrangement would be similar. It should be noticed that the water and other drums are circular in shape; the oval arrangement having been discarded as a result of war experience for this and all other water-tube boilers of the upright-tube type. These circular drums are made large enough for a man to enter the water drums,

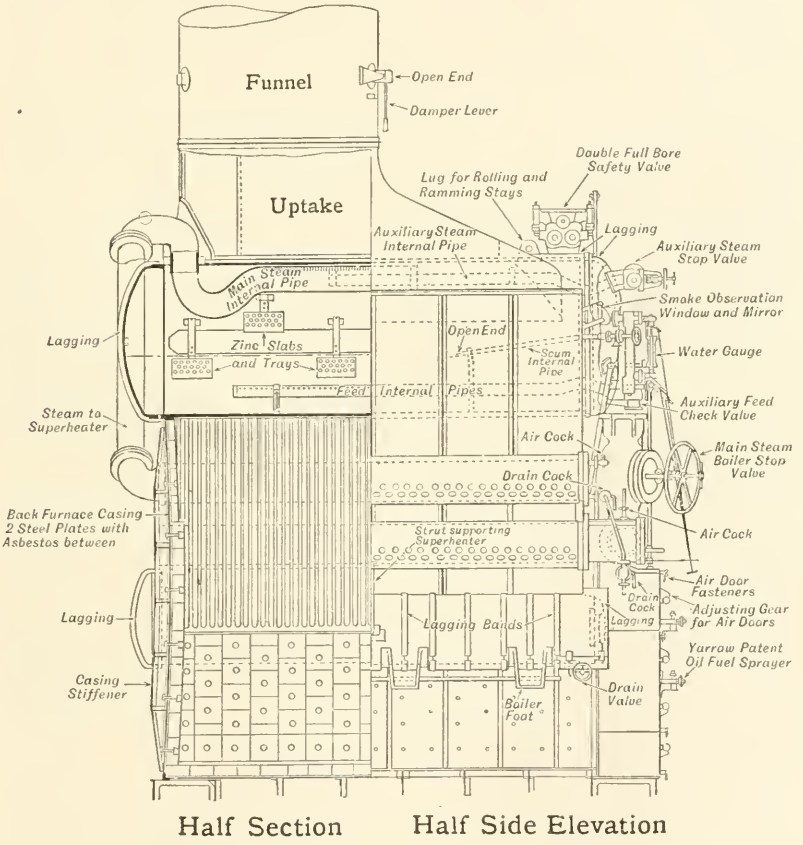


FIG. 46.—Yarrow Water-tube Boiler, with Superheater.

but the superheater drums are smaller, and in the Yarrow arrangement the tube holes are arranged so that an expanding apparatus can be introduced from the outside of the drums through a double row of holes, which are subsequently plugged as shown.

Two internal feed pipes, one main and one auxiliary, are fitted, and in general the water passes down the tubes more remote from the

fire, and upwards through those nearest the fire, accommodating itself to the circulation necessary. The steam passes through an internal steam pipe to an external pipe in connection with the lower drum of the superheater, then passes through the superheater tubes to the upper drum, which is in connection with the main steam valve and pipe. The auxiliary steam does not pass through the superheater. All the drums are fitted with drainage arrangements and the upper superheater drum is fitted with an air cock.

The holes for the reception of the tubes are first drilled through the tube plates, then a harbouring tool is used to drill away a portion of the plate in the wake of the holes, so that each tube shall be in contact only with a right-angled cylindrical surface of moderate length, as shown in Fig. 51. After the tubes are in place they are expanded by a tube roller, in the usual way, to make them tight in their holes, and then the ends are bell-mouthed with a pneumatic hammer. The withdrawal force, or total unbalanced axial tension in each tube, is about 700 lb. for a pressure of 270 lb. per square inch. Some experiments with a tube for the *Triumph* showed that it required a force of 2 tons to stretch the tube about $\frac{1}{64}$-inch, and that withdrawal only took place after stretching the tube $5\frac{5}{8}$ inches with a pull of $13\frac{1}{2}$ tons.

In some of the upright-tube type boilers, the feed water has been pumped into the lower (water) drums, but this system has been abandoned owing to local corrosion from liberation of air and oxygen from the feed water.

The boiler shown in Fig. 46 was tested on shore by Messrs. Yarrow for comparison of oil fuel consumed, water evaporated, steam pressure and temperature of superheat, and on the side not fitted with the superheater for the temperature of the gases at various points during their passage past the boiler tubes.

The results of these trials are shown in the annexed tables A and B, and the curves shown in Fig. 47.

According to Table A, the maximum rate of evaporation was 18 lb. per square foot of heating surface per hour, and 14·6 lb. per 1 lb. of oil fuel burnt, which gives a rate of 1·237 lb. of oil fuel consumed per square foot of heating surface per hour, the superheat being then 93° Fahr.

[TABLE

TABLE A.—TRIALS WITH DAMPER OPEN.

Heating Surface { Large Nest of Generator Tubes = 3247 square feet; Small Nest of Generator Tubes = 2188 square feet; Superheater = 1265 square feet } 6700 square feet total.

On these trials the heating surface is taken as the total heating surface of 6700 square feet.

Steam Pressure, lb. per square inch.	Superheat in degrees Fahrenheit.	Air Pressure, inches of water.	Lb. of Water evaporated per hour.	Lb. of Oil Fuel burnt per hour.	From and at 212 degrees Fahrenheit.		Lb. of Oil Fuel burnt per square foot of heating surface per hour.	Temperature of Feed Water degrees Fahrenheit.	Temperature between Small Nest of Generator Tubes and Superheater, degrees Fahrenheit.	Temperature of Uptake degrees Fahrenheit.	
					Lb. of Water Evaporated per lb. of oil per hour.	Lb. of Water Evaporated per square foot of heating surface per hour				Above Superheater.	Above Large Nest of Generator Tubes.
242	93·5	5·0	94,659	8286	14·6	18·0	1·237	58·0	1121	828	887
243	93·0	3·16	76,021	6454	15·0	14·4	·9635	63·5	926	698	727
243·7	82·5	2·44	68,387	5695	15·2	12·9	·850	63·5	903	685	688
242·8	61·1	1·7	46,041	3630	15·9	8·6	·542	64·0	647	536	551
241·8	31·0	·998	20,059	1540	16·1	3·7	·230	62·2	481	432	448
242·2	20·75	·625	8,478	649	16·1	1·55	·096	63·5	465	409	416

TABLE B.—TRIALS WITH DAMPER SHUT.

Heating Surface { Large Nest of Generator Tubes = 3247 square feet; Small Nest of Generator Tubes = 2188 square feet; Superheater = 1265 square feet } 6700 square feet total.

On these trials the heating surface of boiler is taken as heating surface of Large Nest of Generator Tubes = 3247 square feet.

Steam Pressure, lb. per square inch.	Air Pressure, inches of water.	Lb. of Water evaporated per hour.	Lb. of Oil Fuel burnt per hour	From and at 212 degrees Fahrenheit.		Lb. of Oil Fuel burnt per square foot of heating surface per hour.	Temperature of Feed Water, degrees Fahrenheit.	Temperature of Uptakes, degrees Fahrenheit above Large Nest of Generator Tubes.
				Lb. of Water evaporated per lb. of oil per hour.	Lb. of Water Evaporated per square foot of heating surface per hour.			
242·0	4·85	68,648	6287	13·25	25·66	1·936	61·0	913
242·25	3·97	57,693	5065	13·84	21·6	1·56	60·0	843
242·4	2 491	44,050	3504	15·3	16·5	1·09	60·3	673
242·5	1·46	31,481	2473	15·4	11·75	·76	63·5	603

In Table B, the damper on the superheater side is closed and it is assumed that only the opposite side of the generator is in active operation.

In Table C, shown in form of a curve, the results are extremely interesting, and confirm the calculations made some years previously

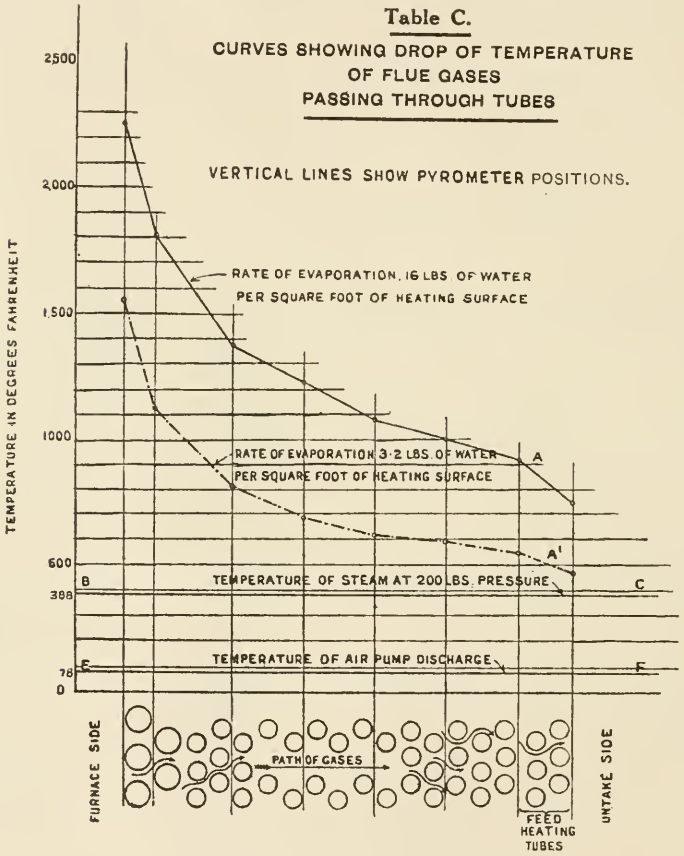


Fig. 47.—Temperature of Flue Gases.

by the author as shown in Chapter XI. In this test, however, the last two rows of tubes are made to act entirely as feed heating surface by fitting a longitudinal partition plate in the upper, or steam chamber, so as to avoid the short-circuiting of the feed. The steepness of the curves at A and A′ show the beneficial action obtained by confining the downward circulation to a definite number of tubes remote from the fire. Thus at A′, just prior to the gases passing the feed heating

tubes the temperature of the gases is about 550 degrees compared with a feed water temperature of 78 degrees, and leaves a large margin for feed heating. If these tubes were filled with steam and water of a temperature of 388 degrees corresponding to 200 lb. pressure the

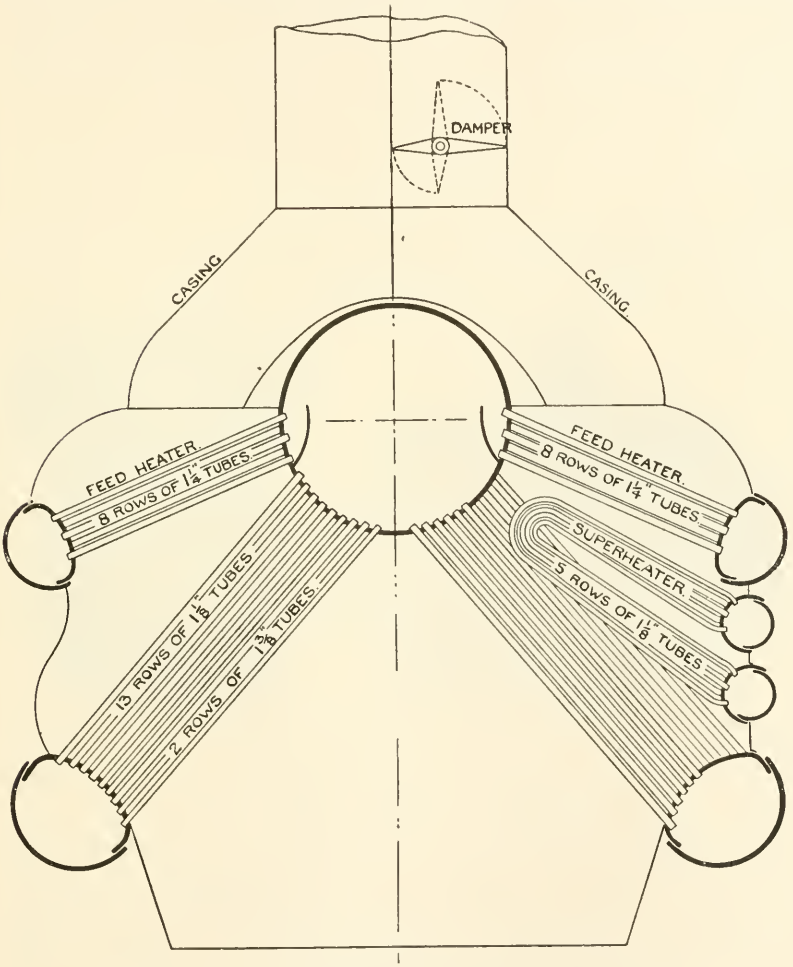


FIG. 48.—Yarrow Boiler fitted with Feed Heater and Superheater.

margin for heating is only 162 degrees, and the probable and possible heating would be very much less.

The results of the tests shown in this curve led Messrs. Yarrow to design the arrangement shown in Fig. 48, combining two feed heaters

and one superheater with the ordinary arrangement of generator tubes. Boilers somewhat similar to this, but with the feed heater tubes horizontal, have been fitted in several warships by Messrs. John Brown, and others by Messrs. Yarrow, but with a superheater on each side. In these arrangements the feed heater tubes are short enough to enable them to be fitted and withdrawn through the steam drum. In some cases the superheater tubes are fitted at right angles to the generator tubes, that is horizontally fore and aft.

Thornycroft Boilers

Figs. 49 and 50.

In the British Navy the type consisting of one upper steam drum and three lower water drums has long been superseded by the types

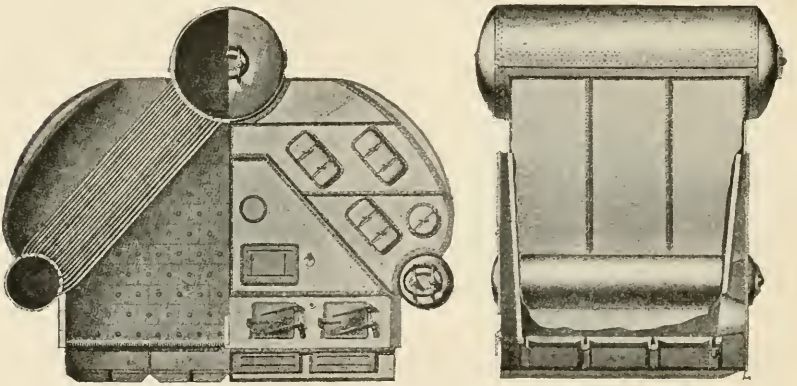


Fig. 49.—Thornycroft Boiler for burning Coal.

shown in Figs. 49 and 50. The general construction is similar to that of other makers except for the curvature of the tubes.

Fig. 49 shows the latest type of Thornycroft boiler arranged to burn coal fuel. It should be noticed that the tubes are curved slightly only at the lower ends, which allows for unequal expansion, but does not interfere with sighting the interior throughout the entire length of the tubes. With this slight curvature the lower barrels can be kept small in diameter, and consequent small weight compared with straight tubes, and a reduction of dead water and inequalities of temperature and racking stress. The lower barrels are cylindrical in accordance with recent practice, and in a number of cases are made of seamless steel, thus doing away with all joints and fear of leakage. The curvature

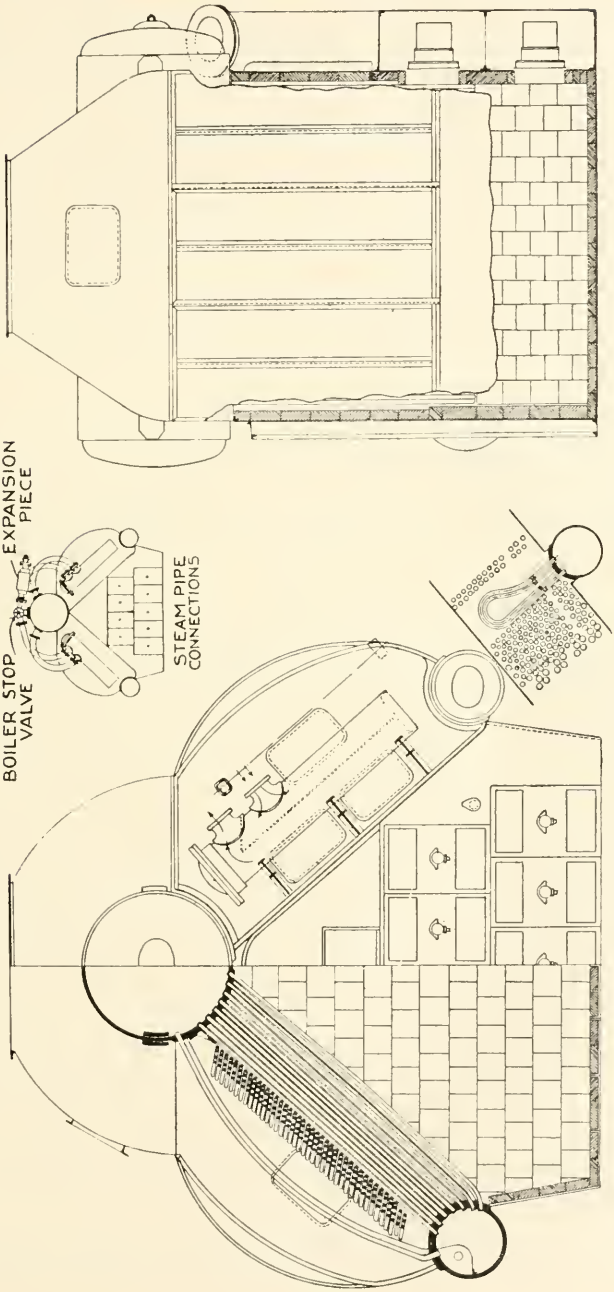


Fig. 50.—Thornycroft Oil-fired Boiler, with Superheater.

of the tubes at the lower ends is the same, and this reduces the number of spares. External cleaning is arranged for by fitting the tubes in straight rows fore and aft, or from front to back, and in straight rows at a cross angle from the furnace to the smoke-box.

Fig. 50 shows the latest arrangement of Thornycroft boiler for oil-fuel burning and superheating to about 100° F. The construction is similar to that of Fig. 49, with the exception of the casing and omission of the firegrate. All parts are made of wrought steel and no castings are necessary. The superheater barrel is large enough for a man to enter for examination purposes, and the superheater can be removed bodily from the boiler without disturbing the generator tubes, etc.

In all types of Thornycroft boilers the fire row tubes are usually made $1\frac{1}{2}$ inches in diameter and the remaining rows $1\frac{1}{8}$ inches diameter externally for mercantile or large vessels.

The top end of the tubes is below the working water level ; in other words, the tubes are drowned, and there is no above water delivery.

Upright Water-tube Boilers—General Remarks

Plugging or Renewing Defective Tubes.—A defective tube in an upright-tube boiler is generally plugged and left until others in the vicinity require similar treatment or cutting out and renewing. A few tubes less do not affect the rate of evaporation materially, and if properly plugged they are quite safe, as shown by practical experience. The plug used is generally a smooth cone made of steel, and inserted in the end of the tube from the inside of the boiler. The pressure inside the boiler keeps the plug in place, and a flange left on the plug prevents the plug from being driven through the tube plate either by the hammer or pressure under steam. Both ends of a defective tube must be plugged, and when the fires are alight the tube burns away, but the part in contact with the water remains intact.

In some cases the renewal of a tube presents some difficulty unless it forms one of the outer rows. Where the tubes are simply expanded into the holes in the tube plate, the ends are now bell-mouthed to prevent their accidental withdrawal. Generally such bell mouths must be cut off before the tube can be withdrawn, and if any other tubes are in the way of the withdrawal these must first be removed. A tube once removed is generally unfit for replacing.

When the tubes, originally straight, become distorted and bent,

considerable discretion should be exercised in making renewals, because of differences caused by expansion under steam and possible drawing or pushing of the tubes on the tube plate holes, and consequent loosening of the ends. Shortness of water may cause the tubes to leak at the upper joints particularly and distort the holes and sometimes the tube plate. In one instance the upper tube plate was distorted about one inch in its length. Examination of the tubes showed them to be in good condition except for leakage, and it was decided to re-roll and expand them without any renewals. This was carried through successfully, thus effecting a great saving in time and expense during the war, when tubes were scarce and time was of the utmost importance. In

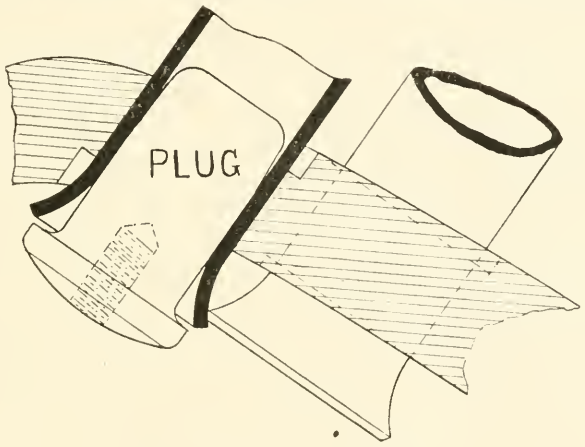


FIG. 51.—Plug for Defective Upright Tube.

this case the leakage had prevented any serious overheating of the metal, and the prompt action of the engineering staff in shutting off the oil-fuel supply had prevented further damage. In another instance, prompt action was not practicable, and the whole of the tubes had to be renewed as well as one of the upper tube plates.

Tubes.—For upright-tube boilers, the diameter of the tubes next the fire is generally about $1\frac{1}{4}$ inches, and thickness about 0·128 inch. The remaining rows are generally about 1 inch diameter, and thickness 0·104 inch.

In larger vessels the diameter was $1\frac{3}{4}$ inch, and thickness 0·1875 for the fire rows and 0·156 for the remainder; but this involved increased weight and space, and smaller diameters have been adopted recently for H.M.S. *Hood*, 144,000 s.h.p.

For example, the increase in diameter of each tube from 1 to $1\frac{3}{4}$ inches increases the water space about two and a half times, while it increases the heating surface only $1\frac{3}{4}$ times for the same efficiency and power. The weight of water alone in the boiler is thus increased by over 40 per cent, and all other weights of material are also increased. A greater weight of water is a possible advantage, but the same advantage, with greater efficiency, may be obtained by increasing the number of rows of 1-inch tubes and thus increasing the heating surface.

Curved tubes are preferable to straight tubes and are probably more durable, because they are not then subjected to continual distortion and unnecessary fatigue of material at every time of raising steam or varying the rate of combustion, especially when oil fuel is used. The curvature necessary is not very great, only about $\frac{1}{4}$ inch in each foot of length, which is sufficient to determine the direction of movement when the tube is heated or cooled. This small curvature does not prevent efficient cleaning and examination.

Water-Walls.—To deflect or baffle the furnace gases some upright-tube boilers had two rows of tubes bent in such a way that the passage between the tubes was entirely blocked for a part, usually nearly the whole of their length. This method is now uncommon, and very little, if any, advantage is obtained by it in practice.

Steam Collector or Drum.—This is generally constructed of two semicircular plates, double riveted together longitudinally with either lap or butt joints. Double butt joints are more usual and are generally specified. The upper plate is of uniform thickness; but the lower one, which forms the tube plate and receives all the upper ends of the tubes, is thicker, except where it is machined down to the thickness of the upper and end plates to form the joints with them. These two plates form a circular shell, which is closed at the ends by two domed or dished plates, flanged as necessary to fit the circular shell. The front plate is riveted to the shell, and presents a rounded face towards the stokehold. The back end is sometimes made with an external joint, for convenience of manufacture and for machine riveting; but the dome should project outwards in all cases.

The necessary boiler mountings are fitted on the front plate, except the steam valves, safety valves and air-cock, which are more conveniently fitted on top of the circular shell. A manhole is fitted as convenient, but generally in the front plate and clear of the boiler mountings.

The steam-chest is fitted internally with the necessary zinc pro-

tectors and internal steam-pipe. In addition, nearly all water-tube boilers are fitted with a feed regulator and float gear, either partly or wholly outside the chest. Boilers with above-water delivery are also fitted with internal separator plates. (See Figs. 53 and 54.)

Water-Chambers.—The construction of the water drums is very similar to that of the steam-chest. Two plates are generally used, one thick and one thin, but the shape of the drum was frequently only partly cylindrical, as shown in figures 44 and 48, but is now always circular. In all cases the part forming the tube plate is thicker, and when the plate is flattened the thickness is variable to withstand distortion. A manhole is usually fitted to each water drum, at the stokehold end, for access to the interior and to the tube ends for cleaning and repair. A drain valve, or blow-down, is generally fitted to each water drum, with which a hose connection can be made for running the boiler water into the reserve tanks or bilges.

The water drums rest on the boiler seatings fitted in connection with the framing of the ship, and when steam is up the generating and downcomer tubes generally support the upper parts of the boiler. Framing in connection with the ship is also fitted to support the steam-chest when cold, and is specially constructed so as not to check the necessary expansion of the boiler, particularly in height. This framing prevents the boiler shifting on its seating when the ship is rolling and pitching.

CHAPTER VIII

WATER-TUBE BOILERS: NEARLY HORIZONTAL TUBE TYPE

Babcock and Wilcox Boiler

Typical Features.—The generating tubes are straight, and inclined at an angle of about 15° with the horizontal; each tube is connected with two headers with wavy sides; and a number of such tubes in connection with a pair of upcast and downcast headers forms an *element*, as shown in Fig. 52. Each downcast header is connected at the top with the steam chest or collector, and at the bottom with the feed collector, which are common to all the elements, 16 to 20 in number, of a single boiler. Each upcast header is connected at the top with the steam chest through a pair of top return tubes.

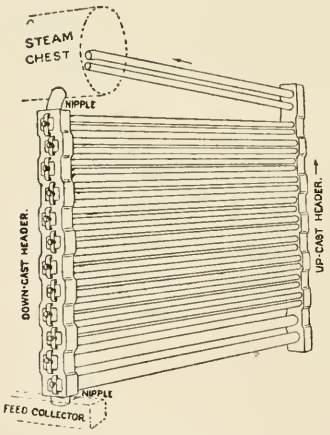


Fig. 52.
Babcock and Wilcox Element.

The tube ends and nipple connections at the top and bottom ends of the downcast headers are made steam-tight in their holes by rolling and expanding them in the usual way. There are two principal designs for naval purposes: one with tubes of $3\frac{3}{16}$ inches diameter throughout, called the medium tube design; and the other containing $3\frac{1}{16}$ inches diameter tubes for the two rows next the fire (as shown in Fig. 52), and $1\frac{3}{16}$ inches diameter tubes for the remaining rows, called the small tube design. The small tubes allow some increase in heating surface (and power) for the weight, and for this reason have been adopted for naval vessels. As shown in Fig. 52, the sight holes in the headers, for fitting,

cleaning, and searching the tubes (to ascertain a clear passage), are square; and for the smaller tubes, one sight hole admits four tubes. This arrangement has also been adopted for naval vessels.

Circulation. — At working height the water level is below the horizontal diameter of the steam collector, and the whole of the generating tubes and all internal parts of the headers and feed collectors below

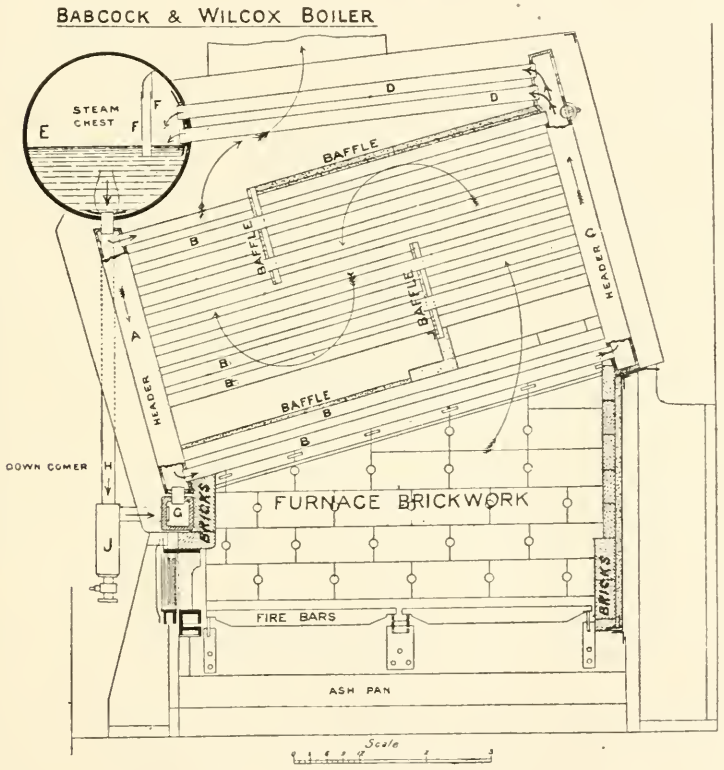


FIG. 53.—Circulation in Babcock Boiler.

this level are therefore submerged, leaving the top of the uptake headers and the top return tubes above the normal working level. The circulation is generally in accord with that described for Fig. 17, but Fig. 53 shows the boiler in fuller detail.

The feed water is pumped into the steam collector E, and is generally automatically controlled when high rates of evaporation are intended. From the steam collector the water passes down the

downtake headers A, and, supplying the generator tubes B, replaces the water evaporated or circulated away into the uptake headers C. The inclination of the tubes sets up a circulation towards their upper ends, and so to the uptake headers ; there the steam tries to rise to the top. From the top of the uptake headers, the steam and some water, impelled by the circulation caused by convection, pass through the top return tubes D into the steam collector E. The top return tubes deliver their contents above the working level in a nearly horizontal direction, but slightly downward ; the steam and water impinge on dash-plates F, the steam rises to the upper part, while the water mingles with the feed and other water in the bottom of the steam collector, where it begins a fresh circuit.

The bottom ends of the downtake headers are connected together by a horizontal tube G of square section, called by the makers the *mud-drum*, but which is actually a feed leveller and collector. Extra *downcomers* H with *sediment* boxes J are sometimes fitted to supply this drum with water from the steam collector independently of the downtake headers. Any roll of the ship above 15°, the inclination of the generating tubes, tends to check the circulation, and consequently these tubes are preferably placed in line with the keel, where they are not so much affected by rolling. In this case (always fitted for naval purposes), when the ship rolls the water is prevented by wash-plates from rushing to one end of the steam collector, and thus leaving the tubes at the other end temporarily denuded of water. The extra downcomers ensure some supply to these tubes through the feed collector.

The *wash-plates* are flat plates fitted across the lower part of the steam collector, with their top edge just above the working water level.

Path of the Furnace Gases.—The gases rise from the fire and pass through the passages between the generating tubes, which are in vertical staggered rows, to the space above ; thence they pass to the uptake leading to the funnel, between two other staggered rows of tubes, conveying the upeast steam and water from the headers into the steam collector.

When the small-tube design is used, the baffling among the tubes is occasionally omitted ; but baffling tends to produce an increased efficiency by retarding the flow of the hot gases to the funnel, and is fitted in warships similarly to the medium-tube design. The first baffle is fitted above the second row of tubes, and extends about

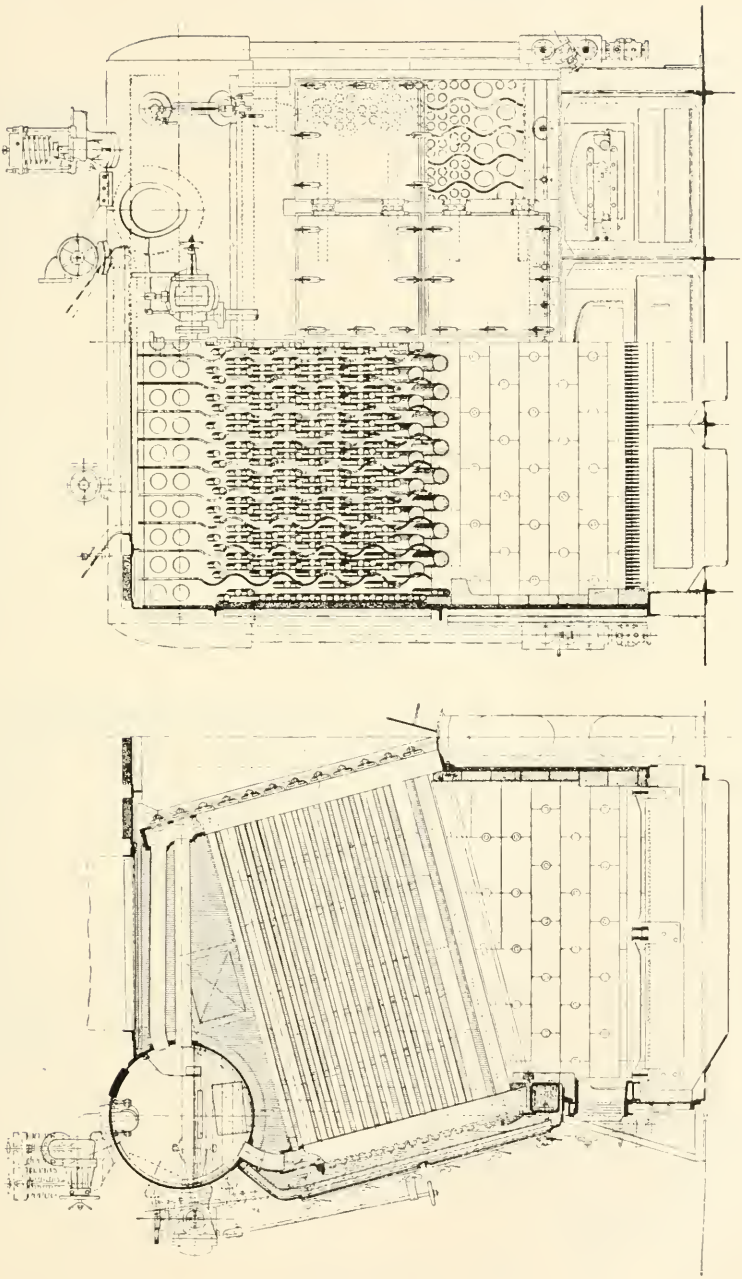


FIG. 54.—Babcock and Wilcox Boiler (small-tube type).

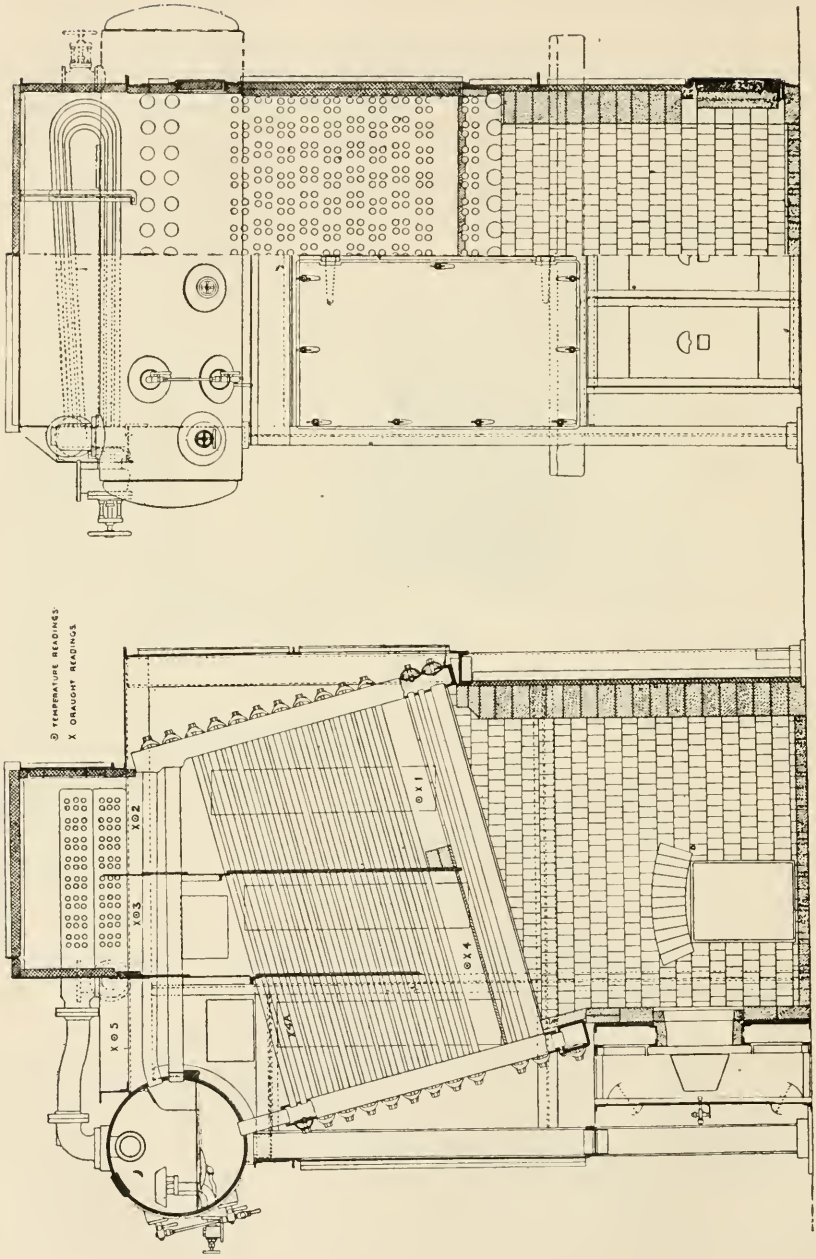


FIG. 55.—Oil-fired Babcock and Wilcox Boiler, with Superheater.

BABCOCK & WILCOX BOILER, WITH SUPERHEATER, FOR BURNING OIL

Boiler heating surface, 2210 sq. ft. Superheater surface, 282 sq. ft.
Combustion chamber, 265 cub. ft.

Date of test	16/9/13	15/9/13	16/9/13	17/9/13	11/9/13	12/9/13
Duration of test—hours	2	2	2	2	3	3
STEAM						
Pressure—lb. per sq. in.	200	202	199	202	197	201
Temperature of superheated steam—°F.	460	461	488	500	506	529
Degrees of superheat—°F.	72	73	100	112	120	141
OIL						
Type of burner	B. & W.	B. & W.	B. & W.	B. & W.	B. & W.	B. & W.
Number of burners used	3	3	3	3	3	3
Oil burned per hour—total lb.	680	827	1,184	1,697	2,362	2,664
,, ,, per burner—lbs.	226·6	276	395	566	787	888
,, ,, per sq. ft. of boiler H.S.—lb.	·308	·374	·536	·768	1·069	1·205
,, ,, per cub. ft. of combustion chambers—lb.	2·566	3·12	4·467	6·40	8·913	10·03
Pressure of oil—lb. per sq. in.	70	103	213	184	146·6	183
Temperature of oil—°F.	219	213	223	211	203	211
WATER						
Temperature of feed—°F.	60	61	62	61·5	62·6	65
Actual total evaporation per hour—lb.	8,275	9,900	14,100	19,500	25,966	29,550
Actual evaporation per hour per sq. ft. of boiler H.S.—lb.	3·74	1·48	6·38	8·82	11·75	13·37
Equivalent evaporation per hour from and at 212° F. per sq. ft. of boiler H.S., excluding superheater effect—lb.	4·54	5·43	7·72	10·68	14·21	16·15
Equivalent evaporation per hour from and at 212° F. per sq. ft. of boiler H.S., including superheater effect—lb.	4·715	5·63	8·11	11·29	15·06	17·27
Actual evaporation per hour per lb. of oil—lb.	12·17	11·97	11·91	11·49	11	11·092
Equivalent evaporation per hour from and at 212° F. per lb. of oil—lb.	15·33	15·06	15·15	14·71	14·1	14·33
Factor of evaporation, including superheater effect	1·26	1·26	1·272	1·28	1·282	1·292
NOTE *The position of Draught, Gauges and Pyrometers will be seen by referring to opposite page.* Draught of water in inches ×1	—	·175	·5	1·4	2·67	3·22
×2	—	·17	·45	1·2	2·24	2·7
×3	—	·13	·4	1·0	1·94	2·3
×4	—	·0	·16	·56	1·23	1·5
×4A	—	—	·0	·187	—	—
×5	− ·15	− ·17	− ·2	− ·3	− ·38	− ·2
Temperature of gases °F. 01	1,258	1,247	1,481	1,547	1,857	1,754
02	633	676	811	979	1,040	1,122
03	450	466	574	660	730	790
04	510	584	496	831	900	937
05	338	367	451	523	586	604
Air pressure in stokehold in inches of water	·2	·55	1·1	3·21	4·7	5·3
Air pressure in air boxes in inches of water	·18	·4	·95	2·22	4·0	4·8
Percentage of heat absorbed by steam in boiler	75·38%	74·28%	73·64%	71·00%	68·15%	68·6%
Percentage of heat absorbed by steam superheater	2·83%	2·86%	3·72%	3·98%	4·08%	4·76%
Calorific value of fuel as fired—B.T.U.	18,921	18,868	18,921	18,948	18,868	18,868
Transmission of heat per sq. ft. of boiler surface per hour—B.T.U.	4,388	5,245	7,128	10,323	13,734	15,604
Transmission of heat per sq. ft. of superheater surface per hour—B.T.U.	1,290	1,580	2,950	4,563	6,445	8,427
SMOKE	Light to moderate.	Very light brown at all times.	Very light brown throughout trial.	Very light brown throughout trial.	1st hour, rather dark brown, fairly thick. 2nd hour, light brown and moderate. 3rd hour, light brown and very slight.	Very light brown at all times.
Analysis of Chimney Gases						
CO_2	12·33%	12%	11·5%	12·45%	12·62%	12·8%
O	4·45%	5%	5·95%	4·76%	4·31%	4·12%
Efficiency of installation	78·21%	77·14%	77·36%	74·98%	72·23%	73·36%

two-thirds of their length from the front end, then rises parallel to the headers about half-way among the upper rows. The gases are deflected downwards by a second baffle, fitted on top of the top rows, and extending from the back end about two-thirds of their length. They then turn upwards, after passing below an extension of the second baffle, which projects downwards and blocks about one-third the passage-way. Thence the gases pass upwards between the rows of top return tubes, and so to the uptake leading to the funnel.

The path of the furnace gases, as described, embodies an efficient principle. The gases surround, when hottest, the parts of the generating tubes and headers which contain water and steam of the highest temperature, and after being cooled during this process they are conducted away towards the cooler ends of the generating tubes and downtake headers. The top return tubes are, however, in a comparatively cool place; but as the temperature is still probably from 200° to 300° in excess of the steam and water inside the tubes, some heating is obtained from the escaping gases. A feed heater placed in this position is more efficient, and has been fitted with this type of boiler in some installations.

Tubes.—In all cases the tubes are expanded by tube rollers into the headers to obtain pressure tightness, and in addition the ends are bell-mouthed, to resist the tendency to withdraw from their holes.

When the tubes are $3\frac{3}{16}$ inches diameter throughout, or, as called by the makers, "the medium-sized tube design," the thickness of the two rows of tubes next the fire is $\frac{9}{32}$ (0·28) inch, of the third and fourth rows $\frac{7}{32}$ (0·22) inch, and the remainder 0·16 inch thick. The thickness of the top return tubes is 0·192 inch for all designs.

For the small-tube design, shown in Figs. 54 and 55, the two rows of tubes next the fire are $3\frac{5}{16}$ inches diameter and $\frac{5}{16}$ inch thick, swelled at the stokehold end to a diameter of 4 inches for a length of about 3 inches. The next four rows are $1\frac{3}{16}$ inches diameter and 0·144 inch thick, and the remaining tubes are of the same diameter, but 0·128 inch thick. All the small tubes are swelled at the stokehold (the withdrawal) end to a diameter of 2 inches for a length of 2 inches.

Steam Collector.—The steam collector is a circular structure placed across the top of the downtake headers, next the stoking platform. The ends of the steam chest are closed by circular dished plates, and the independent downcomers project downwards, near each end. The collector is strengthened by plating, where the top return tubes and the circulation nipples are connected with it.

All the boiler mountings are attached to the steam collector, with the exception of the blow-down cocks, which are connected with the bottom of each sediment box. If independent downcomers and sediment chambers are not fitted, then the blow-down is fitted to the mud-drum directly. The position of the mountings is indicated in Fig. 60 (Chap. IX.), and the casings in Fig. 82.

Nipple Connections of Headers. — The connections of the headers with the steam collector and with the mud-drum are made through short lengths of tube about 4 inches in diameter, and expanded into the holes to obtain pressure-tightness, as shown in the figure. The ends are also bell-mouthed to resist the end pressure, or axial tension.

Cleaning Doors and Closures.—Holes are made in the forged steel headers for fitting the tubes in place. For the larger-sized tubes and connecting nipples, *oval* holes were originally made instead of square, and closed by internal oval doors or closures. The inside face of the header around each hole is machined to a true surface, and on this the joint of each door is made with a jointless asbestos ring. The smaller holes are closed either by a coned plug, with its apex projecting outwards, which is drawn up to its seating by a faced nut, making another joint on the outside, or by a rectangular door opposite the ends of a group of four tubes, as shown in Fig. 52. Provision is made for access to both ends of the tubes by smoke-box doors of the usual pattern. Generally, corrugated plating is used for all parts of the casing and smoke-box doors.

Babcock and Wilcox Boiler, with Superheater.—This arrangement is shown in Fig. 55 for a boiler fitted for burning oil fuel. The baffling, although similar in principle, differs somewhat from the boiler previously described. The results of the test are shown in the annexed tabular statement, which will repay careful study.

The **Belleville Boiler** has ceased to be fitted in H.M. ships for nearly twenty years, and reference does not appear to be necessary in the present edition, but it has been fully dealt with in previous editions. In the French navy it still finds a place, and a notable development is the new arrangement of tubes in the elements. Instead of a single tube arising from the first junction box, two are fitted one immediately above the other. The first junction box at the back end takes one of these tubes and the junction box above it takes the upper one. The same arrangement holds good for all the remaining junction boxes, and in this way the inclination of the tubes to the horizontal is thus

increased to over 4° instead of the $2\frac{1}{4}$ to $2\frac{1}{2}$° obtained by the older arrangement. The upper junction box receives two tubes one above the other to compensate the double system at the bottom junction.

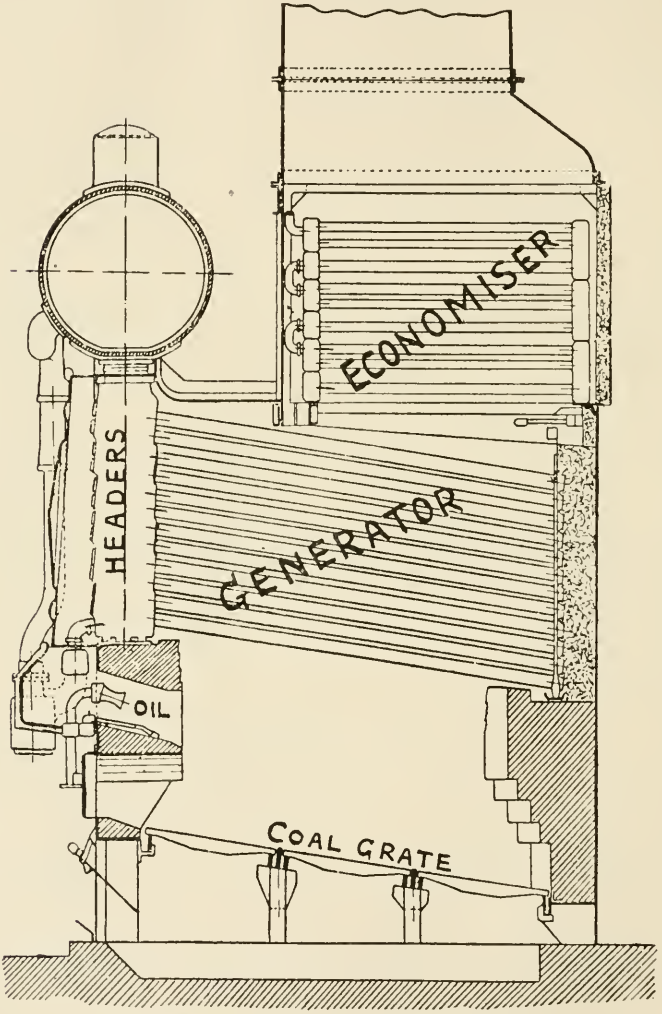


FIG. 56.—The Niclausse Boiler, with Economiser.

The boiler on trial responded to all requirements of both coal and fuel consumption satisfactorily.

The **Niclausse Boiler** is also fitted in the French navy and is

now fitted with an economiser as shown in Fig. 56. The inclination of the tubes to the horizontal is 15° for the generator tubes, and is practically horizontal for the economiser tubes. The feed water is forced in at the top and flows along and downwards to the bottom of the economiser and thus to upper chamber, or steam drum of the generator.

The arrangement of a Niclausse tube bundle is shown in Fig. 57. It consists of an inner and outer tube. Water flows in through the inner tube and out through the space between the two tubes to the upcast header.

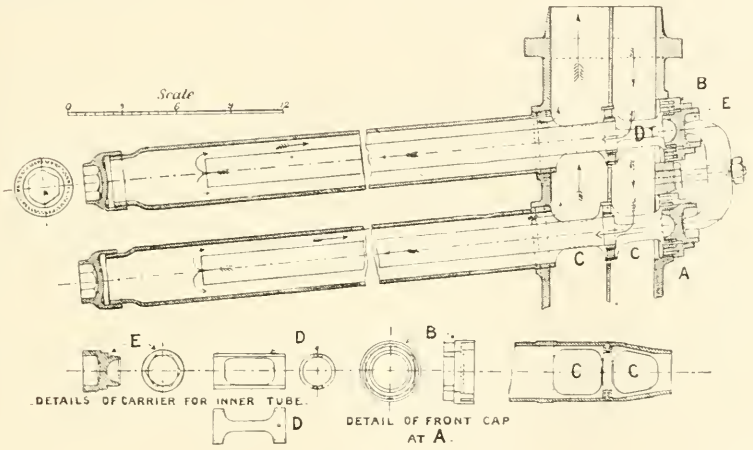


FIG. 57.—Niclausse Tube Bundle.

The **Dürr Boiler** is similar to the Niclausse in principle, but the front headers are made into one only with numerous stays between front and back plates.

CONSTRUCTION AND DESIGN OF WATER-TUBE BOILERS

The almost universal introduction of oil fuel combustion in the British Navy has accentuated the practical standardisation of water-tube types of boiler during the war period. For the larger vessels the types most commonly fitted have larger tubes, generally about $1\frac{3}{4}$ inches in diameter, such as the Babcock and Wilcox small tube type and the Yarrow, Thornycroft and White-Forster large tube types.

For the smaller and generally faster vessels, the boilers are entirely confined to the upright-tube types mentioned in the previous chapter.

With the decreased steam consumptions obtained by turbine

installations on the single and double reduction geared systems, it is practicable to build boilers for the larger vessels with 6000 S.H.P. per unit, and in the smaller vessels, where small weight is of the greatest consequence, up to 9000 S.H.P. per unit. The upright-tube boiler adapts itself very easily, within reasonable limits, to very large units and the necessary large combustion space in connection with such units for oil fuel combustion.

Superheating, generally to about 100° F., is becoming common in all types of warships and also the mercantile marine in combination with turbine installations for coal or oil fuel fired boilers. Higher degrees of superheat are under construction, and an increase to 250-300° is probable in the near future.

Grate area, for coal fired boilers only, is generally based on a coal consumption of 20 lb. per square foot per hour, but for oil fuel plants there is no grate, and the combustion calculations are based on the cubic space allowed for combustion of the oil fuel. In practice it is found that the efficiency of oil fuel combustion shows a considerable falling off above a certain figure, and an allowance of 1 cubic foot of combustion space is now generally allowed for every 10 lb. of fuel to be burnt per hour as a maximum.

Heating surface, on which the output of steam from any boiler is principally and almost entirely dependent, is the most important element in the design of a boiler. In the older types of boiler, and in those of the mercantile coal fired boilers, it was usual to allow from $2\frac{1}{2}$ to 3 square feet of heating surface per I.H.P. per hour, but with the modern oil fired boiler using superheat of about 100° F., and consuming not more than 15 lb. of steam per hour per 1 S.H.P., it is necessary to provide only about 1 square foot of heating surface at full power with a boiler efficiency of over 70 per cent. Without superheat it is usual to allow about 15 per cent more heating surface than stated above.

Rate of Combustion.—From the above remarks it can be deduced that :

(*a*) For coal, the maximum rate of combustion allowed is 20 lb. per square foot per hour for mercantile vessels and natural draught, up to 30 lb. for naval vessels and forced draught, calculated on the grate area.

(*b*) For oil fuel, 10 lb. of fuel per 1 cubic foot of combustion space ; and under good conditions 15 lb. of water are converted, from and at 212° F., into steam for each square foot of heating surface.

Steam Space may be less with high than with low pressure because the total weight of steam contained in the boiler is greater per cubic foot of space. For cylindrical boilers the space allowed is generally between 0·35 and 0·45 cubic feet per I.H.P. at full power, and the water level is then at about one-fourth the diameter of the boiler from the top. For water-tube boilers the steam space is generally much less, partly because of the greater pressures used. The formation of *wet* steam is less liable to take place with a greater proportion of steam space and water surface area, and the space generally might be increased with advantage and a greater steam efficiency obtained. With an internal steam dryer in connection with the internal steam pipe, as shown in Fig. 72, a reduction in steam space and consequently weight and size of the steam drum is practicable with increased efficiency compared with the usual design, which is based on the size of drum necessary to receive the tubes while obtaining a convenient arrangement of the boiler generally and its lower water drums.

The **Thickness of Plating** for the steam and other drums is based on considerations similar to those employed for a cylindrical body or shell of structures subjected to internal pressure :

$$\text{Test pressure} = \left\{\begin{matrix}\text{Thickness of}\\ \text{metal (in inches)}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Strength of metal}\\ \text{in inch-tons}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Efficiency}\\ \text{of joint}\end{matrix}\right\}$$

The Test pressure = $1\frac{1}{2}$ times the working pressure ;

Strength of material = Elastic limit, which in this case, in tons per square inch,

$= \frac{4}{9} \times 28$, for cylindrical boilers (plates *not* exposed to flame)

$= \frac{4}{9} \times 24$, for water-tube boilers with plates exposed to flame.

Efficiency of joint = 75 to 86 per cent, about, but is dependent on kind of joint and riveting employed.

Parts of the boiler not of cylindrical section require special calculation ; and in certain cases it is necessary to fit stays as already mentioned in connection with cylindrical boilers. Instances of methods used in practice are the staying of the large water space of the Dürr boiler which closely follows cylindrical design ; the square or rectangular headers of the Babcock and Niclausse types ; the mud-drum or feed collector of the Babcock ; the feed collector of the Belleville ; and the partially flattened lower tube plates of the water drums of the upright

tube types of the older designs, but which are now always made circular.

Axial Tension in Water-Tubes.—In all hollow bodies subjected to internal fluid pressure there is a pressure equally distributed on each unit of area of interior surface, and this pressure is exerted longitudinally on a tube as well as radially. In the Niclausse and Dürr boilers, where one end of each generating tube is closed, there is an end-pressure or thrust tending to force the tube away from its attachment to the header or other part of the boiler. The particular form or shape of the body to which the tube is attached does not alter the principle, and the end thrust always = pressure × area on which such pressure is exerted. This area always = smallest area of the hole through which the tube attachment is made, which may generally be taken as that corresponding with the exterior diameter of the tube at this line of contact.

In the *Niclausse* boiler the internal pressure is exerted on an area corresponding to the hole through the back tube-plate ; this pressure tends to hold the tube in place. The front closure fitted opposite the end of each tube, being in rigid connection with the tube, is subjected to an internal pressure which acts in the opposite direction and tends to eject the tube. The two forces are nearly balanced, but the end thrust is slightly in favour of ejection, which is prevented by the dog-stay fitted between each pair of tubes. (Fig. 57.)

In the *Dürr* boiler there is no rigid connection between the front closure and the tube, and consequently the end-pressure is principally exerted in the direction required to keep the tube in place on its coned seating.

In the *Babcock and Wilcox* boiler and in all other water-tube boilers in which the tubes are rolled and expanded into headers or other corresponding parts, the end-pressure is exerted on an area equal to the smallest area of the tube-plate hole and tends to withdraw the tube from its attachment. Such possible withdrawal is now generally prevented by making a bell-mouth on the projecting tube end : as shown in Fig. 51 for a Yarrow boiler. Particulars of a withdrawal test are given above.

In all the water-tube boilers described in this book, with the exception of the Niclausse and Dürr types, every tube for all practical purposes must be considered as a stay-tube, and precautions taken accordingly.

Assuming a working pressure of 300 lb. per square inch, the

withdrawal force, or axial tension in the tube, is about the following :—

Belleville, generator tubes,	$5''$	diameter,	5892 lb.	
,, ,, ,,	$4\frac{1}{2}''$	,,	4770 ,,	
,, ,, ,,	$4''$	,,	3770 ,,	
,, economiser tubes	$2\frac{3}{4}''$	,,	3564 lb.	pressure, about 600 lb. per square inch.
,, ,, ,,	$2''$	,,	1884 ,,	
Babcock, boiler generally	$4''$	,,	3770 ,,	
,, small tubes	$1\frac{3}{16}''$	,,	800 ,,	
Yarrow boiler	$1\frac{3}{4}''$	,,	723 ,,	

and others in proportion.

Stays may be considered as a necessary evil and should be avoided wherever practicable in the design of a boiler. To allow easy access for cleaning and examination they must be pitched at proper distances from each other, and particularly in cylindrical boilers they should be, as well as the tubes which also act as stays, in vertical rows. The bottoms of many combustion-chambers of modern cylindrical boilers are now made semicircular to avoid the otherwise necessary staying and at the same time give a better allowance for expansion and contraction. The rules relating to the staying of flat surfaces in the Board of Trade, British Corporation, Lloyd's, and other regulations should be studied ; being based on sound experience combined with theoretical calculation, it is usual to follow some previous known successful design to check any tendency to inefficiency.

The maximum stress allowable on stays at test pressure is dependent on the quality of the material, as in all cases of stress, and in recent practice is limited to—

18,000 lb. per square inch of section for an effective diameter of $1\frac{1}{2}$ inches and over ;

16,000 lb. per square inch of section for an effective diameter of less than $1\frac{1}{2}$ inches.

The effective diameter is the smallest diameter at the *bottom* of the thread or at any smaller diameter of the stay. Rivets or bolts used for securing stays must be at least 25 per cent stronger than the stays.

The Board of Trade and other Corporations allow 9000 lb. per square inch of net section for solid stays, but in some instances the working stress allowed is less than 8000 lb. per square inch. It is therefore necessary to refer to the rules for any particular case and class of material. Iron is frequently used in place of steel in mercantile

practice both for stays and tubes used as stays, and in this case only about 7000 to 8000 lb. is allowed per 1 square inch of area.

In the design of boilers and other parts of the machinery previous practice is used as a guide. It is necessary to know the I.H.P. or S.H.P. as the case may be, and the probable steam consumption per hour. From these data the Grate Area or Combustion Space (according to whether coal or oil fuel), the Heating Surface, Steam Space and other requirements can be calculated, and a general outline of the dimensions and weight necessary sketched out for more close investigation and eventual design. Cheapness, and frequently efficiency combined with cheapness, can be obtained by adhering to standard-sized plates, tubes and furnaces, with their corresponding rivets, stays, and other material. Unusual sizes are expensive, and sometimes the material is less homogeneous than those commonly made, and therefore less reliable. In these matters it is necessary to be conservative, and to consider any suggested improvement from many points of view.

CHAPTER IX

BOILER FITTINGS AND MOUNTINGS

Furnace Fittings.—A general idea of the firegrate, which is the most important of the furnace fittings, can be gathered from the various figures of boilers in Chapters VI., VII., and VIII.

For a circular furnace, such as fitted in tank boilers, the fire bars are in two or three tiers, and extend, in naval practice, from just inside the fire door to near the beginning of the combustion-chamber; but for the mercantile marine, the back end is shortened by 1 to $1\frac{1}{2}$ feet. Near the fire door the grate is somewhat above the horizontal diameter of the furnace, and is inclined downwards as it recedes, and at about its mid-length is very nearly central. The firegrate is inclined to obtain a better supply of air throughout, while by keeping it as low as possible a larger space remains above the fire for combustion. The air entering below the fire passes through the ashpit door, and is diffused into the fire from between the bars until at the back end only a moderate amount, commensurate with the contracted area, is admitted for combustion. The largest volume of the furnace gases passes over the fire at the back end, and the greatest area is allowed for it. Thus the area of opening for the supply of air and the space above the fire are the maximum amounts within the space allowed by the diameter of the furnace.

The *fire bars* rest on the bearer bars, which are placed across the furnace below the nearly butting ends of the tiers of fire bars. The ends of the bearer bars rest on lugs riveted to the sides of the furnace, and they can be lifted out after the fire bars are removed. The front and back ends of the fire bars rest on *dead plates*—so called because they give no life to the fire, no air being supplied through them. Sufficient side play for the fire bars, and end play for both fire and bearer bars, must be allowed, to prevent any distortion of the furnace and its fittings by their expansion.

The back end is faced with *brickwork* to prevent the dead plate and its supports from burning away. Sufficient area—about one-fifth that of the firegrate—is allowed over the brickwork for the passage of the furnace gases through the throat of the furnace; an arch is sometimes used to protect the riveted joint of the upper part with the combustion chamber.

The *furnace door* of cylindrical boilers is fitted in a wrought-iron or forged steel frame, nearly semicircular in shape, and flanged to cover the projecting end of the furnace, to which it is usually secured by a few bolts screwed through it radially to the furnace; the lower part, which forms the dead plate, rests on lugs fitted to the sides of the furnace. The door is made of dished plate, and fitted with an inside guard plate, secured at a distance of about 2 inches from it. The plate next the stokehold is perforated with a number of small holes, or slits, from $\frac{1}{2}$ to $\frac{3}{4}$ inch in diameter or width. The plate next the fire has an unperforated surface, and allows a sufficient margin for the air to pass, from the holes, round its edge into the space above the fire. A sliding door, or louvre, is fitted to shut off the air-supply through the holes when not required.

The door swings on a horizontal hinge, and can be clipped either open or shut as required. Many fire doors are fitted on the self-closing swing-gate principle as shown in Figs. 44 and 49; that is, the lower pivot of the hinge is brought away from its vertical position below the top pivot, both sideways from the door and outwards from the boiler. There are many varieties, but the one described above is perhaps the simplest and best. The principal requirements are: (*a*) proper supply of air above the fire—about one-fourth might be supplied in this way; (*b*) the fire door should be capable of being locked open in a sea-way, and also be self-closing when required; (*c*) the interior plate or plates should be easily renewable, and should prevent the actual door from warping or burning away.

Fig. 58 shows a section through the door and frame of a Yarrow boiler, and a front elevation of the furnace door with its surroundings removed. Air is admitted above and below the frame, and through an opening with a shutter in the middle of the door.

The *ashpit door* of cylindrical boilers is almost semicircular in shape, and entirely covers the opening below the dead plate. It is generally swung on horizontal hinges resting in lugs fitted at the front of the furnace, and when opened the lower part enters the furnace at the bottom. A rack is provided to keep the door in any required

position ; this arrangement is used for regulating the air-supply below the fire. The hinges are placed above the centre of gravity, so that the door closes naturally when not held open. Handles are fitted and the side lugs arranged so that the door can be lifted away, when required, for pricking the fire, raking out the ashes, etc. A removable bar is fitted across the mouth of the ashpit for use with the pricker, and is known as the *pricker bar*.

Automatically *self-closing ashpit doors* are generally fitted to boilers worked under very high rates of combustion, as in destroyers. The

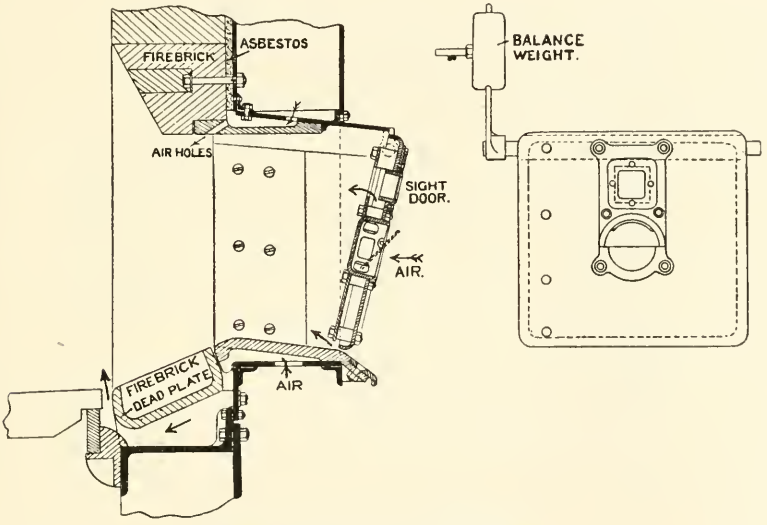


FIG. 58.—Furnace Door and Fittings.

door is slung by three or four links of chain to a pair of gallows inside the casing ; and the relative positions are such that when the pressure inside the ashpit is greater than in the stokehold, the door closes and is pressed against the inner face around the opening. Unless the forced draught is unnecessarily great, the ordinary door opens an insufficient amount, and consequently a larger one is sometimes fitted inside the front of an airtight casing built out from the ashpit ; this permits a larger opening, and thus increases the air-supply with the same air-pressure.

Ash-pans and Water Service.—The ash-pan is sometimes used for corrugated furnaces. It then consists of a thin plate, generally zinced by the hot process, bent to cover the lower part of the furnace

below the fire ; the edges are lipped, or fitted with angle steels which are lipped, to fit into the corrugations, and to prevent ashes and moisture from getting into the space between them and the furnace. The ash-pan facilitates the withdrawal of ashes, and its smooth surface presents less frictional resistance to the flow of air to the fire. Their continued efficiency is doubtful ; they easily buckle from the heat of the fire, when the ashes and moisture have free access below them, and conduce to the corrosion of the furnace.

With water-tube boilers the ash-pan is necessary to prevent the

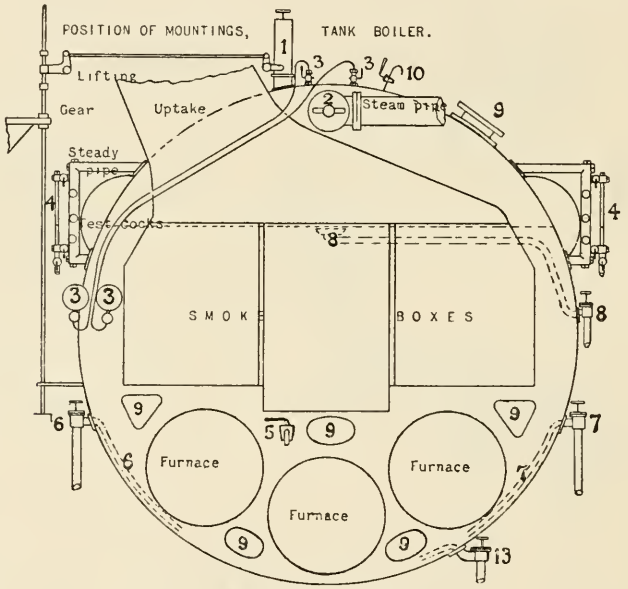


Fig. 59.—Mountings on Cylindrical Boiler.

ashes falling into the bilge. For high rates of forcing a little water is run into the bottom of the ash-pan, and prevents the overheating of the bars by immediately cooling the ashes, and thus conduces to continued efficiency. The water is supplied by two pipes, one on each side of the furnace, which are perforated with a lot of small holes in such a way that the jets of water strike the lower edges of the fire bars below the whole surface of the grate. The pipes are provided with shut off cocks, and these are only opened when actually required for short intervals ; a depth of 1 or 2 inches is generally sufficient for all ordinary purposes.

For *oil fuel*, the ash-pan is bricked over in a water-tube boiler, and

then forms the floor of the combustion space and water service is not required.

Position of Boiler Mountings.—In Figs. 59 and 60 the positions of the mountings usually fitted to a marine boiler are shown for cylindrical (tank) and Babcock and Wilcox (water-tube) boilers respectively. Similar mountings are seen in Figs. 46, 54 and 55, for upright water-tube boilers. The reference to the mountings shown in Figs. 59 and 60 is as below :

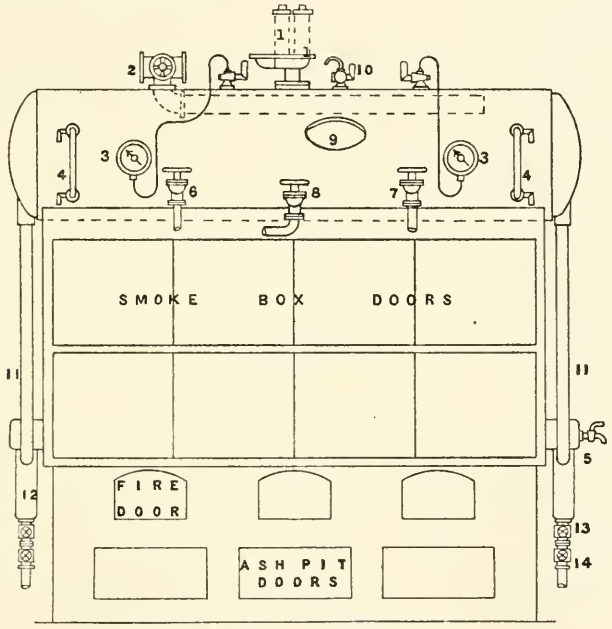


FIG. 60.—Mountings on Water-Tube Boiler.

1. Safety valves, at least two to each boiler; spring loaded with spindles vertical.
2. Steam valve, self-closing type, with spindle horizontal; connected with an internal steam pipe.
3. Pressure gauges, one graduated to rather more than the test pressure.
4. Water gauges; one glass tube type, and generally one plate glass—at least two. Water test cocks or metal water gauge on steady pipe.
5. Hydrometer cock, for testing density of water.
6. Auxiliary feed valve, on left.
7. Main feed valve, on right; if out of reach, gearing is fitted to control each feed valve.
8. Brine or scum valve, with internal scum pan just below the working water-level.
9. Manholes, and doors.
10. Air cock.
11. Downcomer pipes.
12. Sediment chamber; a drain or running down valve is fitted to one sediment box of each boiler.
13, 14. Blow-down cocks.

The construction and other particulars are detailed under their various headings. All valve boxes and gauge mountings are made with spigots projecting into the boiler.

Water Gauges.—The water-level in the boiler is shown by means of a glass gauge fitted between two cocks, the upper one of which is in communication with the steam space and the lower with the water space of the boiler. The gauge glass is generally made in the form of a tube, as shown in Fig. 61, about $\frac{5}{8}$ inch external diameter, and about 14 to 18 inches long; but as pressures have increased, other methods of fitting the glass are also adopted. The gauge for tank boilers is preferably attached to a steady pipe (see Fig. 59) with large openings, not less than 2 inches in diameter, so that there is no danger of their becoming choked, and thus showing a false height of water. Care should be taken not to make the connections with the boiler near any current of either steam or water, because the level may be affected by a slight difference of pressure; for each $\frac{1}{10}$ lb. the level is altered by about $2\frac{1}{4}$ inches. All boilers should have two sets of water gauge mountings, and double-ended boilers are generally fitted with three. The lowest water-level in tank boilers, shown by the gauge, should be at least 4 inches *above* the highest point of the heating surface.

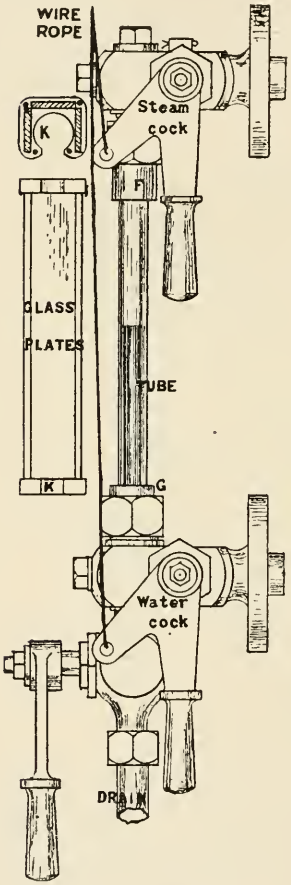


FIG. 61.—Water Gauge.

Three cocks are fitted, as shown in Figs. 61 and 63, to each water gauge: the top, or steam, cock connects the top of the glass gauge with the steam space; the bottom, or water, cock connects with the water space; and the drain cock is fitted below and opens connection when required with the bilge. The gauge should be tested and blown through occasionally. To clear the steam cock, open the drain and close the water cock; steam should then blow freely through the

drain, and the water-level disappear from the gauge. To clear the water cock, shut the steam cock and open the drain; water should come freely through the drain, but the water-level may not be affected in the gauge; most probably the level will rise from condensation in the upper part of the tube. In normal working the top and bottom cocks are open and the drain is closed; all the handles should hang downwards when in their normal working position, and there is then less probability of derangement.

The glass tubes used for water gauges, and other parts subjected to steam and water pressure, should be fire finished, and not *cut* to the required length; cutting reduces their durability.

Water Gauge Safety Fittings.—The ordinary glass tube fractures very easily, and to protect the men in the stokehold from splinters, a glass protector is frequently fitted outside the tube. In Fig. 61 the protector is shown removed to the left of the gauge. It consists of two or three sheets of plate glass, about $\frac{1}{2}$ inch thick, socketed into brass end plates, *K*, *K*, which are kept secured by long brass wires screwed and nutted as required. A section through the protector is also shown in the figure, and it will be noticed that each end plate is cut away for part of its circumference, so that it can be passed over the glass tube, then slipped upwards on the projecting part *F*, below the gland nut, and dropped down over the bottom projection; when the protector is in place, it safely remains until lifted off. The uncovered side of the gauge is usually towards the boiler, away from the stoking platform.

When a tube fractures, the steam escapes through the top, and water through the bottom connection with the boiler. Handles are fitted for closing the cocks from the stokehold platform, the connections for which are shown in the figure. It is preferable to also fit wire ropes for this purpose, and lead them away over rollers to some safe place. When wires are fitted, each cock can be opened or closed independently, but when there is a rigid connection between the two cocks they must be worked together. When steam only is escaping the cocks are generally accessible; but the escape of water is more dangerous, and in some cases an automatic arrangement is fitted to close the lower end of the tube. A small metal ball is supported on a ladle just below the end of the tube, and in such a position that any sudden escape, such as would be caused by a fracture of the glass, throws the ball against the end of the tube and closes the exit; once in position, the pressure inside the boiler holds it there. A similar arrangement is sometimes

fitted to the top connection, but is not so necessary, for the reason already stated.

The glass tube used for containing the water and steam can only be made of a certain strength. If made very thick, the unequal expansion of the interior and exterior surfaces, especially in a strong draught, is very liable to produce fracture, and also the water-level is seen with difficulty.

Window or Plate Water Gauges.—For the high pressures now used, one or both of the usual water gauges, fitted to each boiler, are fitted with glass plates. The arrangement of cocks and steady pipes is the same, but instead of a glass tube, two flat plates of $\frac{7}{16}$ inch thick **Plate Glass,** as shown in Fig. 62, are used to show the water-level. The plates are secured in place by two slotted brass covering plates, front and back, pressed on to packing against a central slotted piece in connection with the mountings used for the top and bottom cocks. The lamp, as in all cases except *reflecting gauges*, should be placed *immediately behind* the gauge.

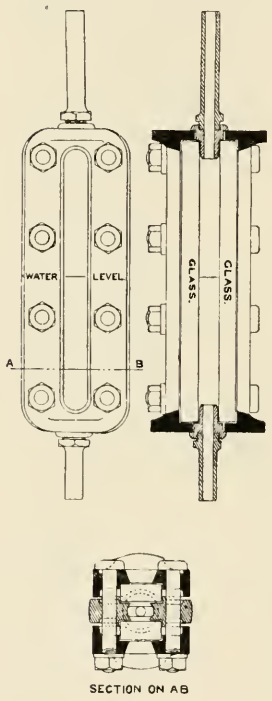


Fig. 62.
Window Water Gauge.

Talc is sometimes used instead of plate glass, and is then about $\frac{1}{32}$ inch thick, but the water-level is not very clearly shown through the ruby-tinted plates.

Combined Glass Tube and Metal Tube Water Gauge.—This arrangement is now being fitted in merchant vessels instead of test cocks, and is designed to give a definite indication of the water-level when for any reason the glass tube is unreliable. It consists of a metal tube in which holes of $\frac{1}{32}$ inch diameter are drilled at intervals in its length, mounted in combination with a glass tube on the usual water-gauge fitting. In the design shown in Fig. 63, and fitted by Messrs. Cammell-Laird, the top and bottom gauge cocks have three-way plugs which enable the metal tube to be switched into connection with the boiler—and the glass tube shut off—in the event of latter being broken or out of order.

The value of the metal tube lies in the readiness with which a small high pressure steam jet can be distinguished from a high pressure water jet, as compared with the unsatisfactory indication given by a boiler test cock.

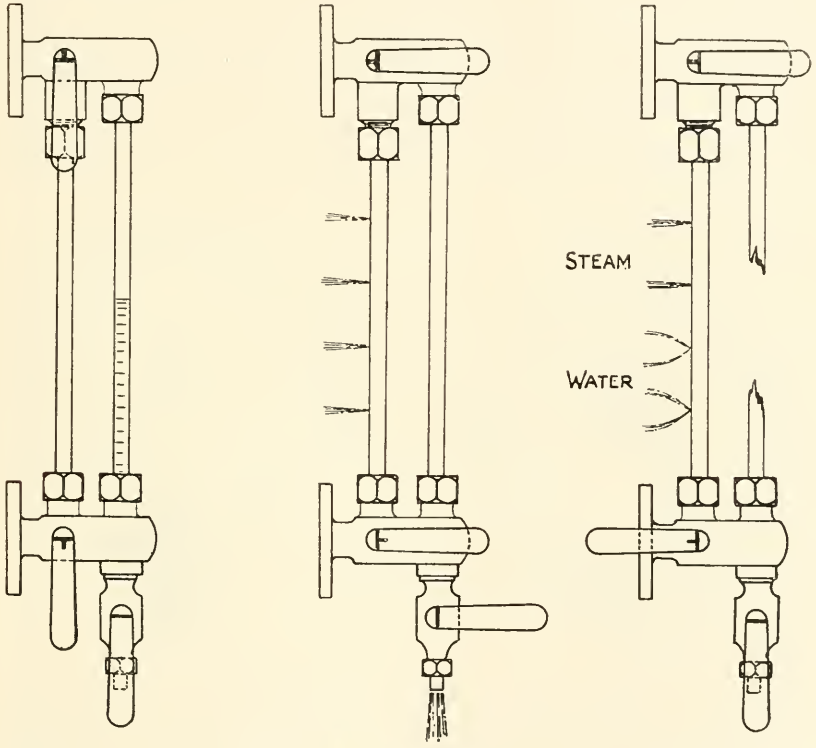


FIG. 63.—Metal Tube Water Test Gauge.

Test Cocks.—Two or three asbestos-packed cocks are fitted at any convenient part of the boiler for testing the accuracy of the water-level. They are generally connected with the steady pipes, if fitted, and one is placed at a level about 2 inches above the highest point of the heating surface, and another about 10 inches above the first. If a third cock is fitted, it is placed about midway between the two.

When a steady pipe is not fitted, and for water tube boilers generally, the cocks are placed about 5 inches above and below the working level, with a third cock, if fitted, midway between them.

Test cocks are not a very reliable guide, especially with high

pressures; but when both water gauges are out of order they are better than nothing for testing the working height of a boiler.

Pressure Gauge.—This instrument is used to show the pressure of steam in the boiler. The front face is graduated to a certain scale, and the pointer indicates the pressure at all times, as shown in Fig. 64. The gauge is fixed at a convenient place, well in sight and near the boiler with which it is in connection; two gauges are fitted to each boiler, one graduated to show about $1\frac{3}{4}$ times, and the other to show about $1\frac{1}{4}$ times, the working pressure.

A pipe, with an asbestos-packed cock or valve at the end nearest

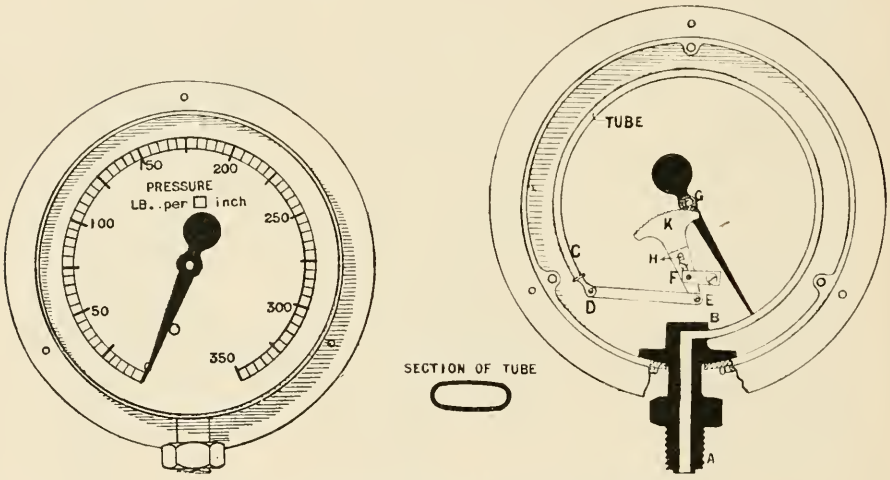


FIG. 64.—Pressure Gauge and Pressure Gauge Mechanism.

the boiler, connects up each gauge with the steam space near its highest part. Another cock is fitted in the connection with the gauge, and can also be used to drain away any water that may accumulate in the pipe. It is generally arranged either that the drain must be open, or that the gauge is in connection through the cock with the boiler; in other words, that the gauge cannot be entirely shut off without some indication from the drain. The cock next the boiler is generally kept open by some safe means, so that it cannot be shut by accident or neglect.

The mechanism, shown in Fig. 64, consists of a bent tube, *BC*, generally of oval section, which is internally connected with the external pipe *A*, and thus with the steam in the boiler. The end, *B*, of the tube is fixed. The other end, *C*, is connected by a rod, *DE*, with the short end of a lever, *EFK*, which has a quadrant, *K*, at its other end.

The cogs or teeth on the quadrant engage in a pinion wheel, *G*, which is attached to the spindle actuating the pointer. The bent tube under an internal pressure greater than that of the atmosphere tends to straighten itself; this is due to the greater length and area of the outer part of the bend of the tube, which produces a greater *total* pressure outwards. If the internal pressure is less than that of the atmosphere, the pressure of the atmosphere tends to bend it in still further, so that the gauge can register pressures both above and below that of the atmosphere. The proportions of the lever *EFK* can be slightly varied by means of the slot and screw shown at *H*.

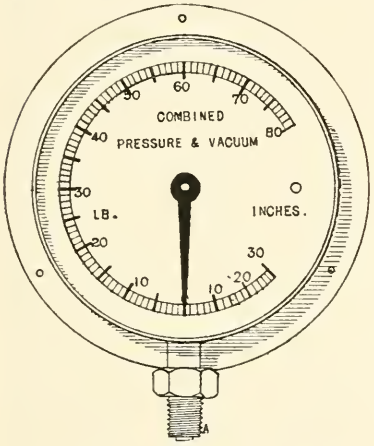


FIG. 65.—Combined Pressure and Vacuum Gauge.

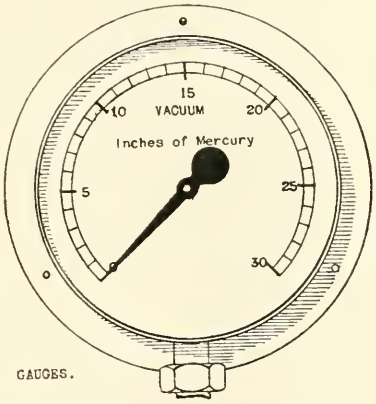


FIG. 66.—Vacuum Gauge.

The *zero* of the pressure gauge is that of normal atmospheric pressure (14·7 lb. per 1 square inch), and a stop pin is fitted to prevent the pointer falling below this point, or being carried right round the gauge, when it would indicate a much lower pressure than that actually present.

Combined Pressure and Vacuum Gauge.—For certain purposes, but not for boilers, a gauge which shows pressures both above and below the atmosphere is used. The zero point is still that of atmospheric pressure; pressures above are shown in pounds per square inch, but the vacuum is shown in inches corresponding with the mercury barometer, as shown in Fig. 65. These gauges are used for the engine receiver pipes, backs of slide valves, condensers, and, particularly, evaporators.

Vacuum Gauge.—The vacuum gauge (Fig. 66) is used to show

only pressure *below* that of the atmosphere; the zero shown is that of atmospheric pressure as on other gauges. At 30 inches of vacuum, as shown by the gauge, the pressure corresponds to 14·7 lb. per square inch *below* atmospheric. In round numbers, 2 inches of vacuum represent an absence of 1 lb. pressure below atmospheric, and so on for each 2 inches; thus a vacuum in the condenser of 26 inches means that the pressure is about 13 lb. below atmospheric or the zero of the pressure gauge.

The internal mechanism is similar for pressure, combined pressure and vacuum, and vacuum gauges, as shown in Fig. 64.

Air-Pressure Gauge.—This instrument is used to show, in inches of water, the difference of pressure in two places, and is a form of water barometer. It consists of a bent glass tube of U-shape, as shown in Fig. 67, one end of which is connected internally with the outer air, and the other is open to the stokehold or connected by tubing with the place required. Coloured water is run into the tube to a convenient height, and when both ends are subject to the same pressure, the level of the water is the same in each leg. When from any cause there is a preponderance of pressure, as in a closed-stokehold, the water-level is depressed on that side, and there is a corresponding rise on the other. The difference of level which can be measured by the movable scale shown is the difference of pressure measured in inches of water. If, on the other hand, one end is connected with the interior of the funnel when steaming, and the other with the atmosphere, the water is depressed on the side open to the atmosphere, because the pressure in the funnel is less, due to the lighter products of combustion which have displaced the atmosphere. The difference of pressure is still registered, but as it is less than atmospheric, it can be termed vacuum.

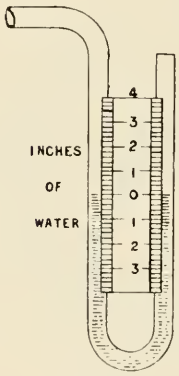


FIG. 67.
Air-pressure Gauge.

The height of the water barometer is about 34 feet, and that of mercury about 30 inches, consequently the latter is less suitable for small differences. The water barometer gives an accuracy at least twelve times as great; thus half an inch on the water gauge would only show about $\frac{1}{25}$ inch on the mercury barometer, which is hardly perceptible.

Safety Valve.—In Fig. 68, one of a pair of safety valves is shown

in sectional elevation as fitted to a marine boiler. It is fitted into the valve box, which is bolted to the boiler at or near the highest point of the steam space. For marine purposes the valve is loaded by springs of spiral form and square section, which are compressed the required amount, so that the valve lifts when the pressure inside the boiler is greater than the load on the springs. The valve is free to move on the spindle to a slight extent, so that if the spindle is not exactly in line with the valve, the latter remains firmly on its seating. In the sketch (Figs. 68 and 68A) the valve is shown connected with the spindle by a cotter, which is a good fit in the spindle, but is allowed some slight play in the socket on the top of the valve. The pressure exerted by the springs is directed centrally on the valves. The top end of the spindle is fitted with a T-handle, so that the valve can be moved round on its seating. This is useful for removing grit or dirt by grinding it away from the face. When such movement is required, and the valve is under a high pressure from the springs, which is not compensated by the pressure inside the boiler, it is usual to partially relieve the pressure thus exerted on the valve face by using the *safety valve lifting gear* outlined in Fig. 59.

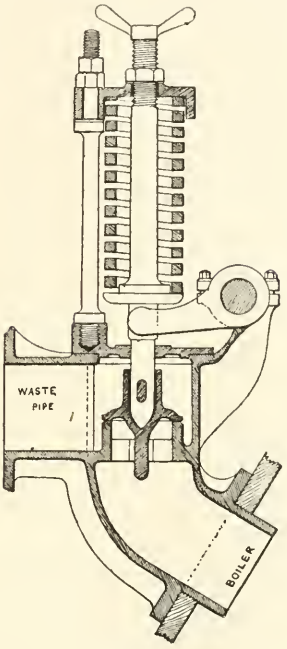


FIG 68.

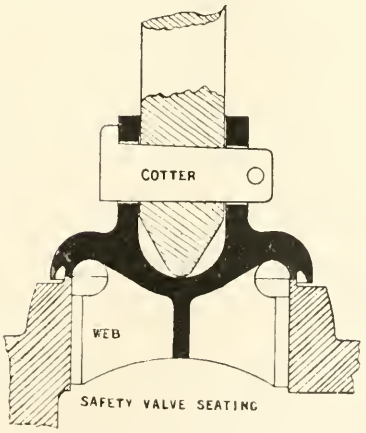


FIG. 68A.

FIGS. 68 and 68A.—Safety Valve and Seating.

High Lift Safety Valve.—The lift of an ordinary spring loaded safety valve at the 10 per cent accumulation test will be 10 per cent

of the working compression of the spring: and where the compression of the spring at the boiler working pressure is the minimum on which the Board of Trade area rule of one-fourth the diameter equals the lift, this lift at the 10 per cent accumulation load will be only one-fortieth the diameter of the valve. The pressure at the base of the waste steam pipe, above the valve, has the effect of still further limiting this small opening. In large unit water tube boilers the restricted area through the valves is insufficient to keep the accumulation within ordinary limits, and in some destroyers a high lift safety valve has been fitted to give the necessary lift.

This arrangement makes use of the pressure which accumulates above the valve, and the pressure is made to act upon a balance piston

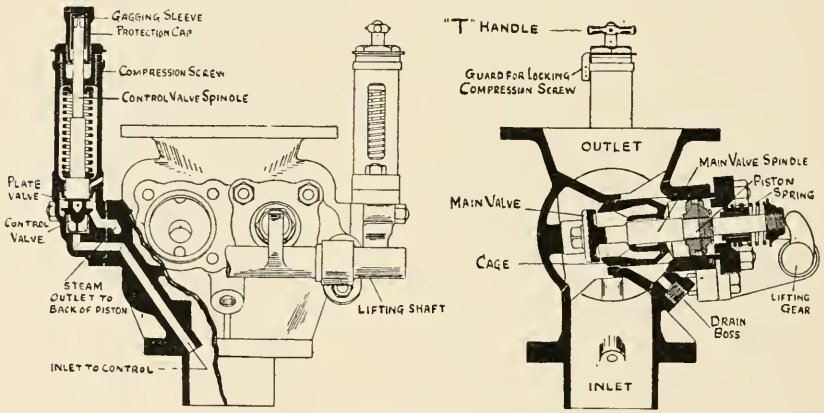


FIG. 69.—Cockburn Full-Bore Safety Valve.

situated on the valve spindle, in such a way as to augment the pressure under the valve and increase the area of opening.

Cockburn Full-Bore Safety Valve.—In this arrangement there are two valves—the control valve which limits the pressure, and the main valve which is operated on by the control valve and thus relieves the excess pressure.

Fig. 69 shows two illustrations of the valves and fittings. Steam from the boiler exerts a pressure on the under side of the $1\frac{1}{4}$-inch control valve; when a predetermined pressure, arranged by the compression of the spring, is reached this valve lifts and steam passes into the chamber directly above. Small holes through the casing surrounding the enlargement of the spindle carry away the pressure in the chamber to the atmosphere and cause the plate valve to close against its seating,

thus keeping the control valve open, and the pressure accumulates sufficiently to pass through the outlet shown to the back of a piston fitted on the main valve spindle, and thus opens the main valve against the pressure exerted on it by the boiler steam. On the boiler pressure dropping, the control valve shuts down and the enlarged part of its spindle carries the plate valve from its seating, when steam again escapes to the atmosphere, which causes the main valve to close, the main valve closing spring assisting in this action. Each main valve has its corresponding control valve.

This arrangement reduces weight, as much as two tons compared with the old type in a 24,000 S.H.P. destroyer ; takes up considerably less space ; reduces leakage from distortion which was very prevalent ; and is simple to refit and examine. It should be noticed particularly that the boiler pressure tends to keep the valve tight on its seating, and that the greater the pressure the more tightly it is pressed to its seating. The actual safety valve is the control valve, which is of very small diameter, and consequently subject to little distortion, and therefore more reliable in every way. This arrangement is usually fitted in boilers of large power, and particularly those using oil fuel.

Steam Valve.—The steam in the boiler communicates with the steam pipe leading to the engines by means of a stop valve, which is usually self-closing ; that is, when the pressure inside the boiler is suddenly released, and is then less than that in the pipe, the valve should close owing to the difference of pressure. By the above means the collapse of one of a group of boilers is isolated from the remainder, and only the steam contained in it is allowed to escape through the fracture. A self-closing valve is shown in Fig. 70 ; the spindle *A* is horizontal, and made in one piece with the valve, so that it can be turned on its seating by the external T-handle. The amount of opening is regulated by the wheel *D* and hollow screwed spindle ; and after setting the opening, which is generally shown by an index *N*, the valve must be pulled back against the collar *F* by means of the inner spindle and T-handle. When the pressure is greater inside the boiler than in the steam pipe, the valve opens itself, as it is then pressed towards the collar ; but if the resultant pressure is in the opposite direction the valve remains closed. Great care should be taken that no undue force is used to open it, and that when it is supposed to be open the spindle is actually withdrawn to butt against the collar. The packing of the gland spindle is important, and generally alternate layers of elastic core and asbestos are used, well smeared with mineral

oil and black-lead ; but in some cases other packings of special character are used. The area of opening through the valve is generally about one-eighth greater than the steam pipe used for carrying away the steam, but in naval practice is frequently less.

The ordinary type of flat valve seating is used for boiler stop valves ; but for high pressures for *bulkhead* and other steam *valves*, a special construction, invented by Messrs. Bevis and Gibson, is sometimes used with two seatings, the inner of which is flexible, as shown in Fig. 70. The inner seating, when the valve is closed, butts on a flexible metal ring and makes a surface joint with it. The flexible

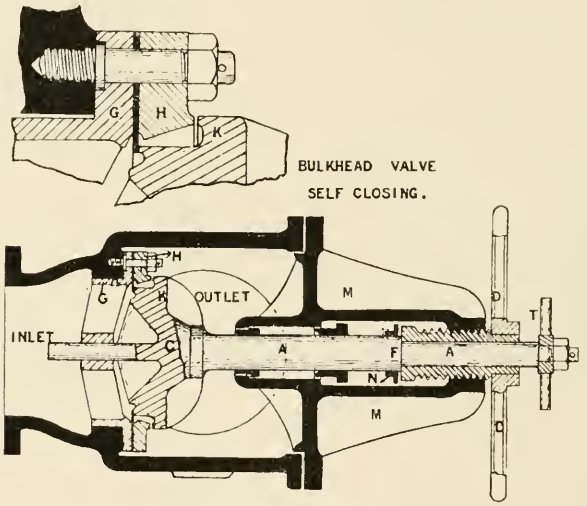


Fig. 70.—Self-closing Steam Valve.

ring is held between the flange *G* and the ring *H*. *H* is suitably made for the seating *K* of the back valve face, which is of the ordinary flat seated type, and, being solid, this seating prevents the valve being driven home too far on the flexible seating and so distorting it. Because the pressure always tends to press the flexible seating against the valve face, steam tightness is obtained with proper fitting. A later type with only one seating, of the flexible type, is shown in Fig. 71.

Internal Steam Pipe.—The stop-valve box is generally made with a spigot, projecting into the boiler through the hole near which its flange is bolted, with which the internal steam pipe is connected. This pipe is generally made of steel or gun metal, and runs horizontally along the upper part of the steam space, as shown in Fig. 46 ; slits

extending about one-third of the circumference are cut on its top side, and the total passage area is 3 to 20 per cent in excess of that through the valve. The object of the internal steam pipe is to prevent the passage of large bodies of water into the external steam pipe, and so to the engines. It acts as a separator inside the boiler; it remits the danger of violent priming, and generally acts as a preventative. Whenever the boiler is opened for examination, or cleaning, the openings are inspected to see that they are clear, and, if necessary, the pipe is removed for the purpose.

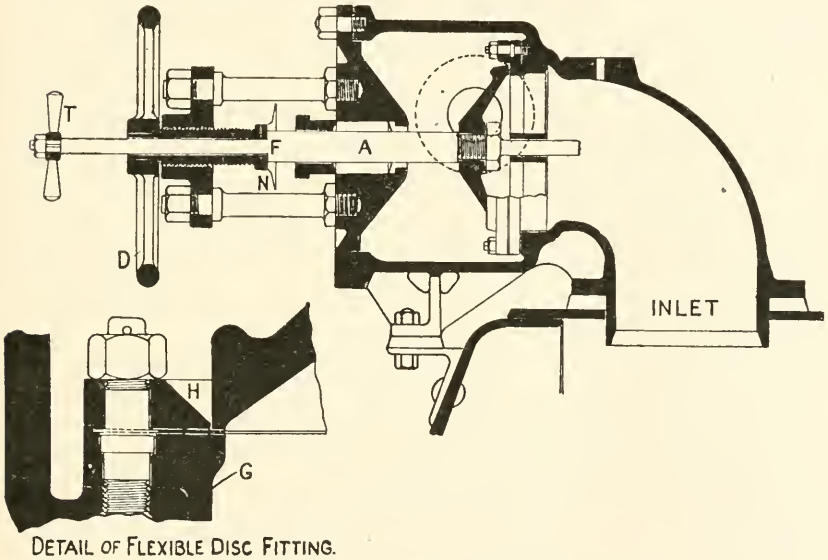


FIG. 71.—Boiler Steam Self-closing Valve. (Bevis-Gibson.)

An auxiliary steam valve is generally fitted of the same construction as the main valve, and is occasionally fitted to take steam from the main internal steam pipe; but a separate fitting is preferable, and more usual.

Prevention of Priming. — The internal steam pipe, fitted with slits of a total area slightly exceeding the pipe area, is not sufficient in itself to prevent priming, but it mitigates priming under good conditions of clean and fresh feed water. A restriction of outlet area through the valve so as to obtain a steam velocity of about 250 feet per *second* also assists materially in preventing priming, but these methods, neither singly nor combined, are successful when the feed water contains any

priming mixture such as sea water, soda, dirt, or other matter either as a mechanical mixture or as a solution. Lime appears to produce no effect either in producing or reducing priming.

Priming, more or less, occurs at all times in all descriptions of boilers, and occasionally makes its presence visible in most inconvenient fashion, and frequently accounts for the great variety of efficiencies obtained from the same steam plant. War service has called particular attention to it, but it has been known as the bugbear of marine steam engineering from the first introduction of steam navigation. Mr. Alexander Richardson, M.P., in his interesting chapter in Brassey's *Naval Annual*, 1919, refers to it as follows:

"Leaky condenser tubes, or 'condenseritis' as it was called in the Service, was perhaps the most prevailing trouble. This led to priming in the boilers with minor difficulties in the turbines. The attention given in recent years to the improvement in non-ferrous metals is having, and will continue to have, its effect in removing this condenser trouble, but leaky tubes were not always the cause of priming, and an instance may be mentioned to show how unforeseen difficulties may arise. In one ship the exhaust steam pipe from the forward capstans was smashed by shell when the ship was in action, and as the cruiser forged ahead at a great speed the sea water flooded the pipe into the exhaust system, and thus into the feed supply of the boilers. The engineer, of course, knew nothing of the mishap, and soon the speed of the cruiser dropped to 10 knots."

A very large drop in speed could be authenticated in many cases from access of sea water to the feed water by other means, such as leaky feed tanks. Many cases could be cited extending over the author's forty years' experience. The only excuse for its continuance is that priming has been looked upon as inevitable under certain conditions.

The simplest and only successful method so far invented, which can and does prevent priming water from leaving the boiler, is shown in Fig. 72. It is fitted in connection with the valve spigot and, if fitted, with the internal steam pipe. A spiral helix, or fixed propeller, providing a passage area of slightly greater area than the passage through the valve, is fitted in a suitably arranged circular pocket, and the steam is thus made to whirl during its passage through the **dryer**, in which the heavier particles of wet steam and water are flung by centrifugal force into the cavity formed by the pocket, and from thence they are continuously driven back into the boiler through a **tangential drain**.

The dry steam, thus cleared of all moisture and wetness, continues its way to the boiler steam valve and the engine.

The tangential drain is made one-tenth the diameter of the internal steam pipe or spigot, and is calculated to deal, if necessary, with at least 50 per cent by weight of water in the steam from the boiler.

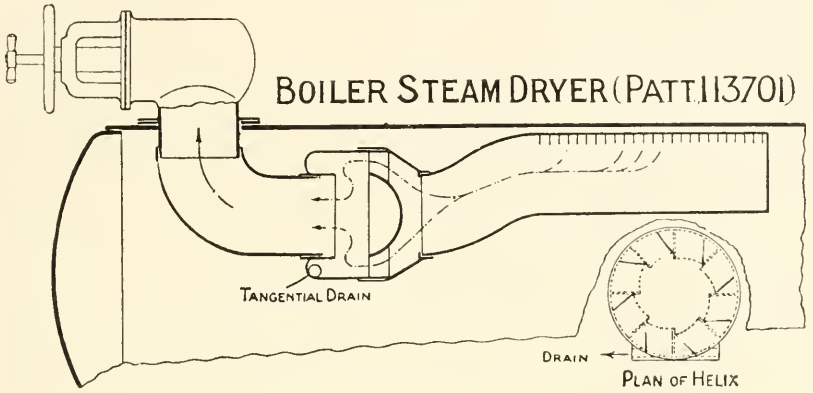


FIG. 72A.—Typical Arrangement.

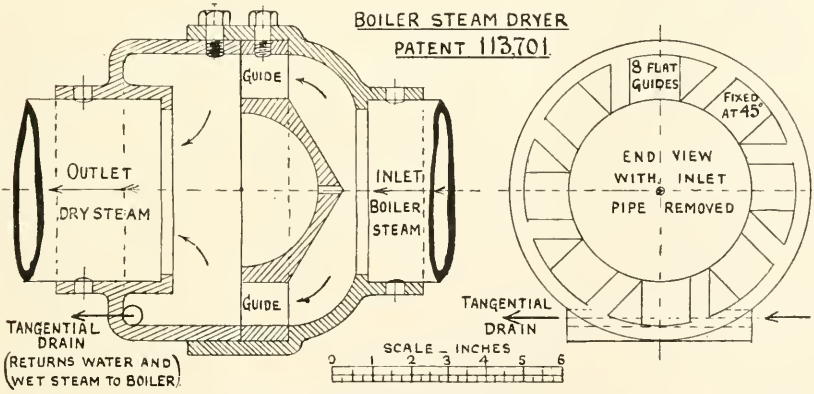


FIG. 72B.—Boiler Steam Dryer (Patent 113,701).

The drain is tangential to the largest diameter of the pocket, so that drainage is continuous whether the drain hole is pointed upwards or downwards or in any direction; in practice, the drain hole is fixed horizontally, as shown in the sketches, and thus serves to keep the pipe drained when the valve is closed. The helix may be right or left handed, and to prevent mistakes the drain hole is made straight through

at a tangent to the pocket and left open at both ends, as shown in the sketch ; it is thus fool proof, and assists the internal circulation of the steam in the boiler.

The action is entirely automatic and continuous, and there is no loss of heat or energy such as occurs in all external separators fitted in engine-rooms, in connection with steam pipes which are consequently wasteful and expensive. With this steam dryer the full pressure of the steam can be carried at the H.P. receiver, and the boiler steam valves opened to their full extent. The agents for this patent, which has also been adopted for steam pipe drainage in connection with whistles, syrens, and engines generally, are Messrs. Ward & Crichton, Chapel Walks, Liverpool.

Air Cock.—An asbestos-packed cock is fitted at the highest part of the boiler for the exit of any vapour or air either when the boiler is being lighted up, or being filled to the crown with water for preservation when not required for steaming. Care should be taken that the cock cannot open from vibration—a catch is usually fitted for this purpose—and that it is closed immediately steam begins to blow through it, because it may otherwise become inaccessible.

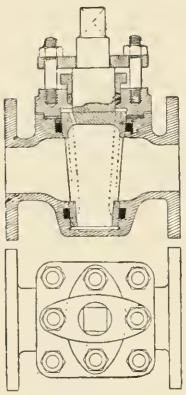


Fig. 73.
Asbestos-packed Cock.

Asbestos-packed Cocks.—For high pressures, valves are always fitted, if possible, in preference to cocks ; but where a full and unrestricted opening is required valves are not always suitable, and cocks are used for indicators, separator blow-downs, boiler blow-downs and brines, drains from cylinders and steam pipes, water and pressure gauges, water test cocks, hydrometer and air cocks. In the merchant service, cocks are also frequently used for regulating the boiler feed and other purposes.

When cocks are used, and are subject to steam or water pressure, they are generally asbestos packed, as shown in Fig. 73. In Fig. 73 the plug does not touch the metal surface, but moves on a non-metallic bearing of specially prepared asbestos packing, which is caulked into grooves in the shell and vulcanised. This packing is shown in black section, and in the shell by dotted lines. The cone-shaped plug tends to be ejected by the steam or other pressure to which it is subject ; to balance this pressure a small hole is drilled through the top end, as shown. Apart from this balancing, the plug is held in place by a spring holding-down plate fitted to bear

on a collar of the plug spindle. Below this plate a gland is fitted and packed to keep the spindle pressure-tight.

For water-gauge cocks and other parts where it is possible to fit them, it is preferable to fit the handle at the small end of the plug-cone.

Feed Valves.—Usually two feed valves are fitted to each boiler, one in connection with the main and one with the auxiliary feed pump. When looking at the boiler, the main feed valve should be on the right and the auxiliary feed valve on the left of the boiler. The valves are always of the non-return type, that is, they close to pressure from the top or boiler side of the valve ; the pressure below the valve must therefore always be in excess of that in the boiler, or the valve closes.

For ordinary hand regulation, gearing is fitted, if required, to work the valve from the stokehold platform. The valve seating is generally raised sufficiently to facilitate repair. Valves with flat seatings reduce hammering, and do not easily become fixed in the openings, and for high pressures they are usually fitted in preference to coned seatings.

The internal feed pipe, if fitted, is connected with a spigot on the valve box, and the water should be directed through it in the same direction as a descending current, which should also surround it, and prevent steam being formed in the pipe. A small hole is generally drilled near any possible pocket or bend, to allow any steam or air to escape ; this reduces the stress on the pipe and work on the pump.

Automatic Feed Regulators.—An automatic feed regulator is fitted to nearly all water-tube boilers in addition to the ordinary valve described above. An automatic apparatus is unnecessary for tank boilers, because with the comparatively very large amount of water contained in them there is very little fluctuation of water level ; whereas if the feed supply were shut off a water-tube boiler for 10 to 15 minutes when under full steam, all the water would be boiled out of it.

Babcock and Wilcox Feed Regulator.—The regulator consists of a hollow cylinder C, Fig. 74, which is rotated on its axis by means of a float and lever placed inside the boiler. The cylinder has ports cut diagonally along its walls and works in a liner having corresponding ports. The liner is fitted in a casing or box which is attached to the boiler shell, and has one branch for a feed check valve V, and another branch for a pipe leading to a non-return feed valve V_1 on the boiler steam drum.

The cylinder can be moved lengthways along the liner by means of

a suitable adjusting screw and spindle, the spindle being connected with the cylinder by ball bearings to reduce friction to a minimum. Attached to the spindle, between the adjusting wheel and the gland, is a lever L and rod, the latter extending to a position from which it can be easily worked by hand from the stokehold platform. This rod and lever are fitted for occasional testing of the freedom of working of the cylinder when getting up steam, or for hand regulation if required at any time.

The position of the cylindrical valve is entirely regulated by the

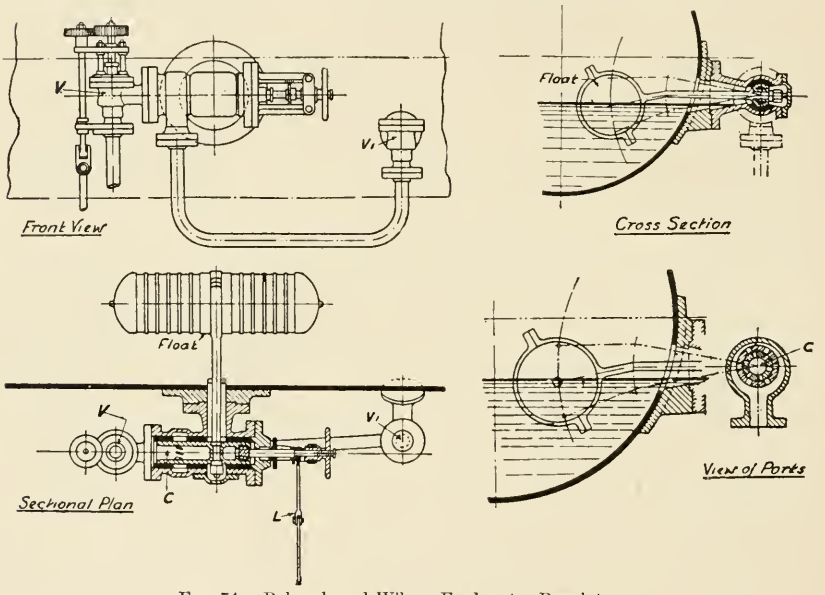


Fig. 74.—Babcock and Wilcox Feed-water Regulator.

float in the boiler, and the height of the water-level can be varied by the hand wheel operating the cylinder lengthways along the liner. At mid position the water-level should show 4 to 6 inches in the gauge glass. The boiler check valve V should be open while steam is being generated and closed when the boiler is not in work. The pressure in the feed pipes should usually be 40 lb. above the working pressure in the boiler when the pump is at work, but when feed heaters and feed pressure filters are fitted the feed pump should have a ratio of steam cylinder area to pump area of 2 to 1. All pipes and fittings in the discharge feed-pipe range should therefore allow for double the boiler working pressure.

White-Forster Feed Regulator.—A sectional elevation of the regulator and an outside elevation are shown in Fig. 75. The float chamber being in connection, through steam and water pipes, with the boiler, the float rises and falls with the water-level in the boiler. Steam being allowed free access, through the top, to the interior of the float, the internal and external pressures on the float are balanced.

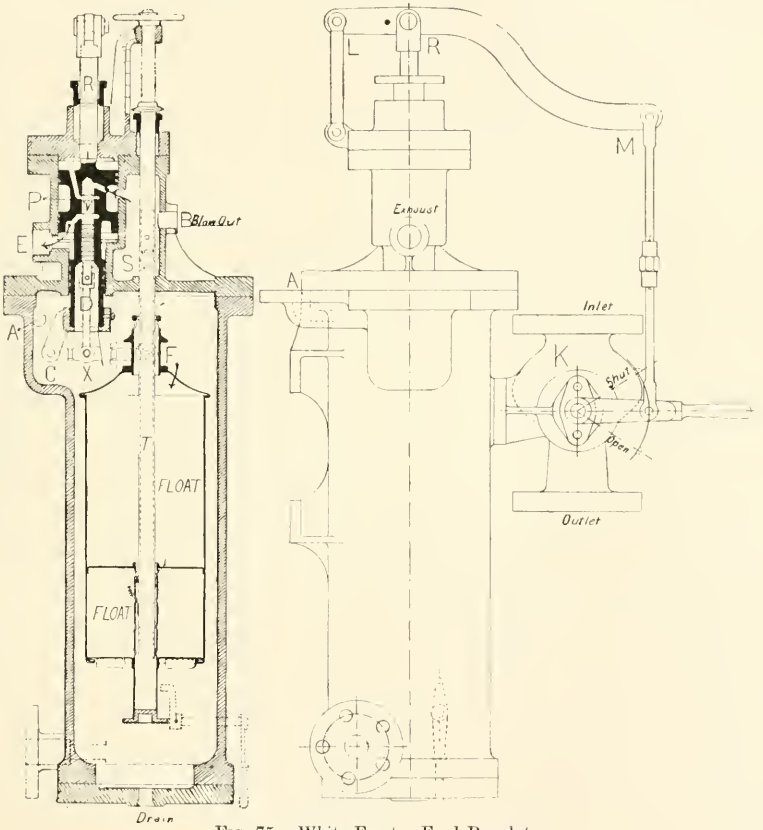


FIG. 75.—White-Forster Feed Regulator.

The float can therefore be made of the lightest material, and consequently requires no weight to balance it.

The movement of the float, moving the lever *FXC*, pivoted at *C*, and the rod *D*, works the valve *V*, which moves inside a double piston, *P*. The piston *P*, shown in black section, copies the movement of the valve *V*, and being connected by a rod, *R*, with the external lever *LRM*, regulates the opening of the feed cock *K*. The piston is

made of two areas, the larger above and the smaller below. The boiler pressure inside the float chamber, always acting on the lower end, tends to force the piston upwards. The steam required for working the upper piston passes downwards from the float chamber through the float, and upwards from the bottom, through an internal pipe, *T*, into the steam space *S*. The exhaust steam passes either into the waste steam pipe or into the auxiliary exhaust.

When the water-level falls the valve *V* is moved downwards, and admits steam from the space *S* to the space above the larger piston. The total pressure on the larger piston overcomes that on the lower piston, and, pulling the external lever downwards, opens the feed cock *K*.

When the water-level rises the valve *V* is moved upwards, and releases the pressure above the larger piston into the exhaust. The constant steam pressure below the lower piston overcomes the *exhaust* pressure on the larger piston, and, pushing the external lever upwards, closes, or tends to close, the feed cock *K*.

The normal working level in the boiler can be adjusted for a few inches, more or less, by allowing a certain depth and weight of water to accumulate inside the float, and thus increase its virtual weight. The increased weight of the float requires an increased depth of water to float it at the original level, and therefore the water-level in the boiler must be correspondingly raised.

Thus, *to raise the normal working level in the boiler*, the steam connection *A* with the float chamber is closed, and the blow-down *B* opened, until sufficient water accumulates inside the chamber to overflow into the float. The tube *T* is then raised the desired amount by the thumb screw, as shown by the index, when the steam connection is again opened to the boiler, and the blow-down is closed as soon as dry steam is emitted from it.

To lower the normal working level in the boiler, the tube is lowered the desired amount, and the blow-down opened until dry steam is emitted from it.

A hand lever is attached to the cock, so that the free movement of the lever and piston can be tested; and another lever is fitted, at the lower part of the float chamber, for testing the free movement of the float and its connections.

Mumford Feed Regulator.—As shown in Fig. 76 there is a float *G*, and float feed regulating box *J*, fitted at about the working water-level outside the boiler. The float box is connected through two

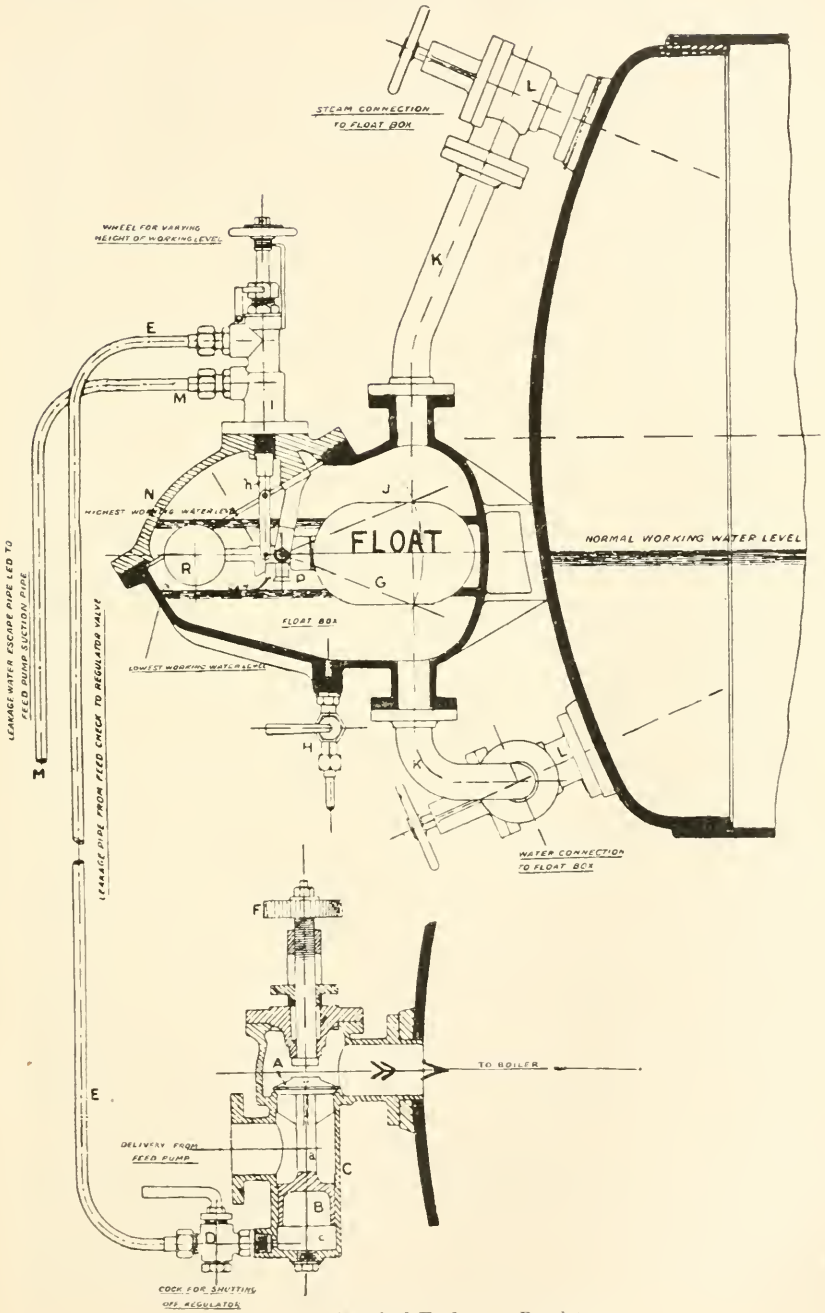


FIG. 76.—Mumford Feed-water Regulator.

valves L and pipes K, with the steam and water spaces. Below the float box and at any convenient position is the feed check valve A, which can be used either for hand F or automatic regulation. The check valve A is of the type which is balanced by fitting a piston B on the spindle below the valve, and the boiler pressure above the valve tends to keep the valve closed at all times. There is a small leakage past the piston which tends to equalise the pressure above and below it, and in connection with the space below the piston is a leakage pipe E leading to the regulating valve which is operated by the float. When the float rises with a rise of water-level it opens the regulating valve and, releasing the pressure below the piston B, causes the check valve to close. When the float falls with a fall of water-level it closes the regulating valve, and, preventing leakage from below the piston, allows the pressure below the check valve to open it against the pressure in the boiler.

The advantages of the Mumford system are that: one size of float regulator and valve can be fitted to any size of boiler, and is independent of the bore of the feed pipe and check valve; all parts are made of gun metal, and therefore not subject or conducive to galvanic action; the working parts are compact, of light weight, and sensitive, and, being fitted outside the boiler, can be easily examined at any time without opening the boiler; a hand regulation can be resorted to by simply closing a cock on the leakage pipe.

Brine Valve.—On nearly all boilers a valve is fitted at some convenient part, and is connected, through a spigot, with an internal pipe, which is fitted with a scum pan slightly below the working water-level. The other side of the valve is connected, through a separate pipe and an asbestos-packed cock, with the sea valve. The brine or surface blow-down is used to remove any floating grease or other matter from the surface of the water in the boiler. The matter thus removed must pass through the brine valve on the boiler, the brine cock, and the sea valve, which may also serve for a group of boilers and for the blow-down. The valve on the boiler is fitted so that the pressure tends to keep it closed; this is used also for feed and blow-down valves.

The brine and blow-down cocks are fitted below the stoking platform, and a handle is provided for each, which cannot be unshipped without closing the cock.

Blow-Down Valve.—The use of the blow-down is to remove any sediment or solid matter from the water by blowing it through

the valve, a separate pipe and a blow-down cock, and the sea connection valve, which is also used for the brine in many cases. The quantity of water, and of the heat thus lost, are given in the chapter on "Evaporators."

For water-tube boilers the blow-down is fitted in addition to a drain valve, with which a hose can be connected for running the water into the bilge, the reserve tanks, or into another boiler by levelling. Inside the boiler the mouth of the pipe which is connected with the drain valve is frequently fitted about a foot or so above the bottom, so that only clean water is thus run off.

Hydrometer or Salinometer.—Sea-water contains by weight about $\frac{1}{32}$ part of solid matter, and if a certain portion of the water be evaporated, the proportion of solid matter remaining is increased. Thus, if two gallons of sea-water are boiled until only one remains, the density or proportion of solid matter will be double that of the original sea-water; in other words, it will contain $\frac{2}{32}$ parts by weight of solid matter. This proportion, or density as it is usually called, is measured by the hydrometer or salinometer, which consists of a weight, float, and graduated stem, as shown in Fig. 77. The density is indicated, on a scale, by the graduation at which the hydrometer floats in the liquid. In the mercantile marine the graduations are sometimes shown in thirty-seconds, thus $\frac{1}{32}$, $\frac{2}{32}$, $\frac{3}{32}$, and so on; but for naval purposes each thirty-second is divided into 10°—thus, the density by the hydrometer of fresh water is *zero*, of sea-water it is 10°, of twice the density of sea-water it is 20°, and so on.

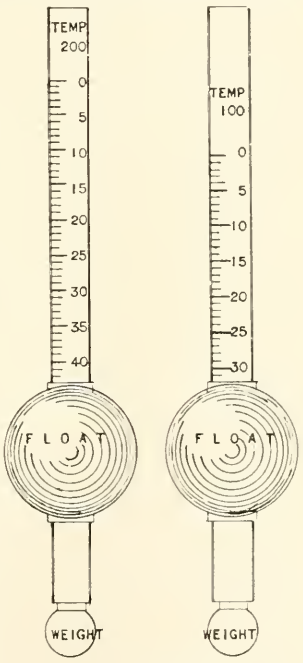


FIG. 77.—Navy Hydrometer.

The density of the water tested depends also on its temperature, and, consequently, the graduations are made to suit the temperature for which it is used. The ordinary naval hydrometer is graduated for two temperatures—that for 100° Fahr. is shown on one side of the stem, and that for 200° is shown on the other, and the temperature is shown near each scale. The two sides are shown in the figure, and it

will be noticed that there is a difference of about 10° between them, due to temperature only.

When the scale is graduated for a temperature of 100° Fahr. it is useful for testing the feed condensed water, and when graduated for 200° Fahr. it is useful for testing the water drawn from a boiler under steam. When great accuracy is required a thermometer should be used for ascertaining the temperature at the time of testing, and the water should be free from steam and vapour.

The boiling-point is also affected by the density of the water, and for each $\frac{1}{32}$ part, or 10°, of solid matter it is about 1·2° Fahr. higher. Thus the boiling-point of ordinary sea-water is 213·2° instead of 212°, which is the boiling-point of fresh water at atmospheric pressure.

Fig. 78.—Sensitive Hydrometer.

Richards' Sensitive Hydrometer.—The comparatively very small amount of water contained in water-tube boilers is proportionately greatly increased in density by the same amount of solid matter pumped into the boiler, and the density may rise very quickly unless it is immediately detected. The ordinary hydrometer, such as described above, is not altogether suitable for the detection of slight increases, and a sensitive hydrometer is now used whose graduations are ten times longer, as shown in Fig. 78. The size of the float is increased and that of the graduated stem decreased, until the relative proportions of the two become suitable for this more minute detection; the principle is just the same in both types. This hydrometer is graduated for a temperature of 100° Fahr.

Chemical Tests of Boiler Feed Water.—With the various testing apparatus supplied by chemists or engineering firms for use on board ship, full instructions are usually given, and it is only necessary here to describe how these tests are made in practice. In all chemical tests the water should be at a temperature of above 60° and below 100° Fahr.

Red litmus turns blue in the presence of an alkali; and blue litmus turns red or tends to bleach in the presence of an acid. Boiler water should always be alkaline and never acid, and in general the litmus shows no change either from red to blue or from blue to red, but a comparison will sometimes give a more definite result; thus, tear a selected piece of litmus in two parts, and immerse one piece and stir

it about in the water under test for some minutes, and then compare any change of tint from the original for either acidity or alkalinity.

Phenolphthalein affords a more delicate test for alkalinity. One drop in 50 cubic centimetres, a large test tube, should turn the water to a magenta-pink colour if a sufficiency of lime is present and the water is alkaline; but if the phenolphthalein is a weak solution or weakened by exposure to air, two or three drops may be necessary to produce the same effect. If no change is effected it denotes acidity, or tendency to acidity, according to the number of drops added and the strength of the solution.

The *silver nitrate test* for chlorine shows the presence or otherwise of salt in the feed water. One drop of the ordinary silver nitrate solution should produce a white cloudy precipitate in a small test tube, 25 cubic centimetres, of feed water if salt is present. The proportion of salt water present can be obtained by next adding (after the cloudy precipitate is formed) one drop of *potassium chromate*, when the cloudy precipitate changes to a yellow colour. If this yellow colour changes to red after adding another drop of silver nitrate solution, the proportion of sea-water present is almost inappreciable, and generally the feed water may be considered in a satisfactory condition; but if it is necessary to add several drops before obtaining the red colour, a search should be made for the cause, and it generally points to a leaky condenser. This test in experienced hands gives a definite result; for example, in an actual test with the usual naval stores and appliances, 25 drops of silver nitrate solution just produced a red colour when the density showed $\frac{1}{10}^{\circ}$ on the sensitive hydrometer, that is, when the proportion of sea-water present in the feed water was only 1 per cent.

Zinc Protectors.—Slabs of rolled zinc, generally $12 \times 6 \times \frac{1}{2}$ inch in thickness, are suspended inside the boiler at any part that is likely to suffer from corrosion. The slabs are connected with the stays by flat mild steel staves secured by bolts, with squares under the heads to prevent them turning in their holes and slacking off. The connections are carefully cleaned and filed bright, so as to obtain proper electrical contact between the stay and the slab.

Ash Shoot.—A circular tube, about 2 feet in diameter, is generally fitted from the level of the main or upper deck, down through the stokehold hatch to the stoking platform. This serves as a passageway for the buckets of ashes hoisted from the stokehold, and they are then emptied overboard through a shoot on the ship's side. The tube also serves as a ventilator in many instances, and then it is continued

upwards until it forms the seating for a cowl, which may be either built of light steel plates or made of canvas. The opening necessary for getting the bucket from the tube is closed by a door when not in use, and, to prevent accidents, it is only of the height necessary for extracting the bucket. At the lower end, a limited height from the stokehold plates prevents any one from getting below the shoot without warning.

When closed-stokehold forced draught is fitted, it is necessary to close the tube to prevent a back draught through it; a moderately air-tight door is therefore fitted both below and above the tube openings, one of which is supposed to be kept closed when the other is open. With this system of forcing, the ash shoot is generally a separate fitting from the ventilating tubes, or only used in part when the fans are not working.

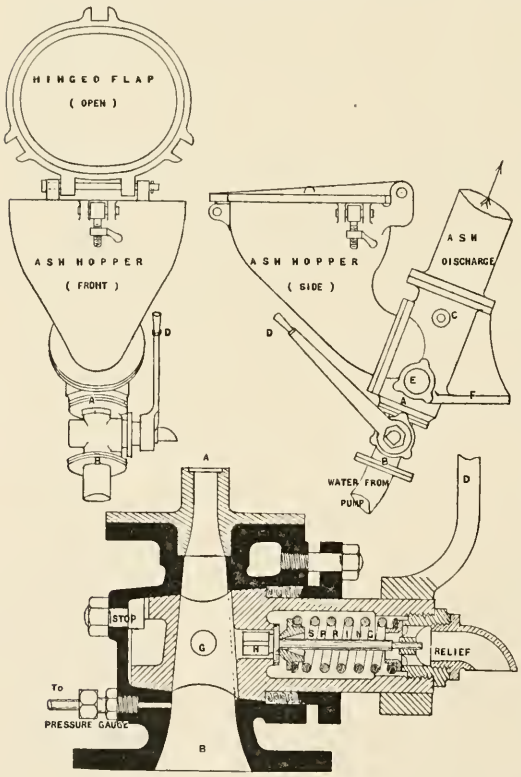


Fig. 79.—See's Ash Ejector.

The ashes are hoisted by a small double-cylinder engine controlled by a differential reversing valve, or, in some ships, by an electrically driven hoist. The bucket, in either instance, is prevented from over-hoist by automatic stopping gear. A gong and voice tube are also generally fitted between the hoisting position and the stokehold.

See's Ash Ejector.—For continuous working at high rates of combustion the ordinary ash hoist is cumbersome, and an excessive number of men are continuously employed in getting up the ashes from below and throwing them overboard. With See's arrangement there is a saving in men, and the stokeholds, which are always too small, can be kept clear of ashes

and accumulation prevented. The arrangement is shown in Fig. 79, and consists of a hopper (just above the stoking platform) into which the ashes are thrown. The hinged flap is closed at first starting.

Water under pressure of over 120 lb. is then admitted by a cock and handle *D* through a pipe *B* and nozzle. The water carries the ashes with it through the discharge pipe, which leads up to the ship's side above the water line. A section (enlarged) through the valve is shown in the lower figure. A pressure-gauge is fitted to show the water-pressure, and also a relief valve in case the pressure gets too high when the shoot is choked. A clearing hole is fitted at *E*. If the discharge pipe is well proportioned and the bend is very gradual, good results are usually obtained, choking is seldom troublesome, and the ashes are thrown well clear of the ship's side. For a 6-inch pipe, about 120 lb. pressure is required on $\frac{3}{4}$-inch nozzle.

Armour Gratings.—Nearly all warships are fitted with armour gratings, as a continuation of the armoured deck or decks, inside all ventilators, hatchways, and funnels, where water-tight doors cannot be fitted, to protect the parts below them as far as possible from shot and shell. In some instances armour hatches or sliding doors are fitted to close above the gratings, for use in extreme urgency; but all of them cannot be safely closed when men are below in machinery compartments, owing to the great heat and lack of sufficient ventilation.

The gratings are somewhat similarly arranged to the firegrate, but the bars are of deeper and rectangular section. A few of them in each hatch are bundled together, and hinged as a whole to form an opening when required. The gratings occupy some of the available area, and, in the interior of the funnel, a net area of opening must be allowed between the gratings for the escape of the funnel gases; the gratings increase the size of the base of the funnel, which is contracted gradually above them to the required net opening.

Fire Extinguishers.—These are fitted to boilers used for strong forced draught in case a sudden loss of water in the boiler, or other accidental occurrence, necessitates putting the fires out. Pipes in connection with the sea, the level of which is always above that of the firegrate, are fitted in front of the boiler (and sometimes at the back also) leading to two or more openings through the casing. The control valve can be worked from some place outside the stokehold, and, when opened, water is sprayed on to the fire. The safety arrangements, such as self-closing fire and ashpit doors, prevent the steam generated in the furnace from gaining access to the stokehold, and the

steam therefore finds its way up the funnel. The fire extinguishers are tried periodically, and always immediately before starting a full-power trial. No fire extinguishers are fitted for boilers entirely fired by liquid fuel.

Sweeping and Searching Tubes.—The best method of external cleaning, so that the efficiency of the heating surface can be continuously maintained while under steam, is to provide for manual appliances for sweeping—using preferably a wire brush. The better classes of upright water-tube boilers are so constructed that a brush can be passed from the stoking platform, between consecutive rows of

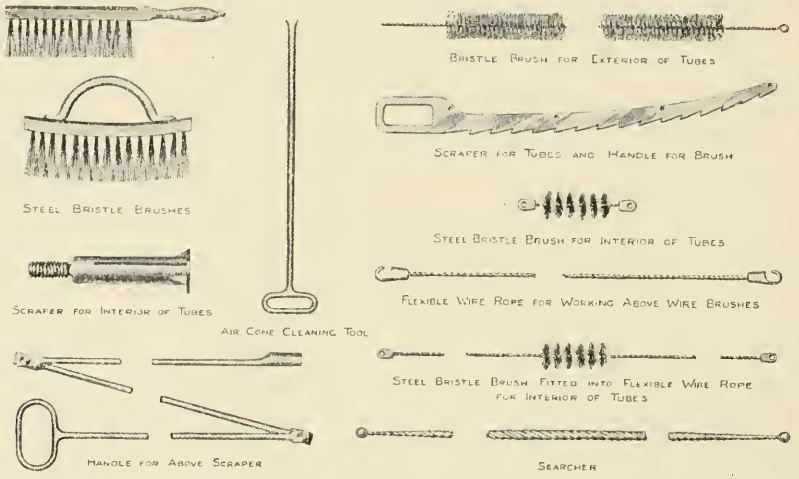

Bristle Brush for Exterior of Tubes
Scraper for Tubes and Handle for Brush
Steel Bristle Brushes
Steel Bristle Brush for Interior of Tubes
Scraper for Interior of Tubes
Flexible Wire Rope for Working Above Wire Brushes
Air Cone Cleaning Tool
Steel Bristle Brush Fitted into Flexible Wire Rope for Interior of Tubes
Handle for Above Scraper
Searcher

Fig. 80.—Thornycroft Cleaning Tools.

tubes, to the back end. The tubes are arranged in straight rows from front to back, and each row has approximately the same curvature.

In 1895 the author used for boilers of the early Thornycroft type first a wire rope, and then washed through with water from the fire-main. For the latter purpose a length of pipe, about one inch in diameter, was connected with the fire-hose at one end, and at the other, with a right-angle bend, a nozzle was formed, so as to just enter the open ends of the tubes in the steam chest. One man manipulated the pipe from inside the steam chest, while another watched the bottom ends of the tubes to see that a full flow of water ran through each tube. This method was successful, and a boiler with nearly a thousand tubes could be completely searched in a few hours without a large expenditure of fresh water, which was always used for the purpose. The tubes are

always searched periodically, and it is usual to search them before steaming at full power.

For boilers of the Babcock and Wilcox type with small doors opposite the ends of the tubes, a scraper or wire brush is commonly used for cleaning the interior. The doors are, of course, previously removed, and, after scraping, each tube must be washed through, because otherwise the scale would lodge in the nearly horizontal tubes and end-pockets.

Boiler Bearers and Anchoring.—A cylindrical boiler rests on two or more bearer frames built above the inner frames of the vessel, and extending from one-fourth to one-sixth around the shell circumference as shown in Fig. 81. Chocks are fitted at front and back to prevent movement of the boiler in these directions, and being secured to the ship, not to the boiler, thus allow sufficient expansion of the latter when under steam. The boiler is allowed free room for expansion circumferentially and upwards, but it is generally anchored loosely, when cold, to the bearers, by bands of flat steel plates, 3 to 6 inches wide, which are secured down to the bearers at B; horizontal stays are fitted near the top as shown. The system illustrated in Fig. 46 is used for water-tube boilers.

Lagging.—Non-conducting material, generally of asbestos or silicate cotton, is fitted around the upper part of the shell of cylindrical boilers and over the back plates, and is carried down to a level just clear of the boiler bearers. The lagging is kept in place by thin sheet galvanised steel plates and angle-steels. The front of the boiler above the smoke boxes and behind the uptake is generally lagged in a similar way, and care should be taken that the uptake and air casing in connection with it should be fitted sufficiently clear of the boiler for periodical examination of the nuts and stay-ends, etc.

Air Casings and Uptake.—The smoke box of a cylindrical boiler is built up of thin plating and jointed on to the boiler, and is kept clear of the tube ends to allow freedom of withdrawal of all tubes without its removal. The lower and generally horizontal parts are provided with small doors, sliding in grooves, for the ready removal of soot when under steam, and when it is not advisable to open the smoke-box doors for the purpose. In front of the casing and opposite the tube ends, doors are fitted, which when open allow the tubes to be swept when under steam, as shown in Fig. 81. The uptakes, shown in Figs. 46, 81, and 82, should be as far as possible without sharp bends so that the gases may have a free exit to the funnel through it.

Around this inner casing, which forms the smoke box and uptake, there is always a second thin casing fixed at about 1 to 2 inches from

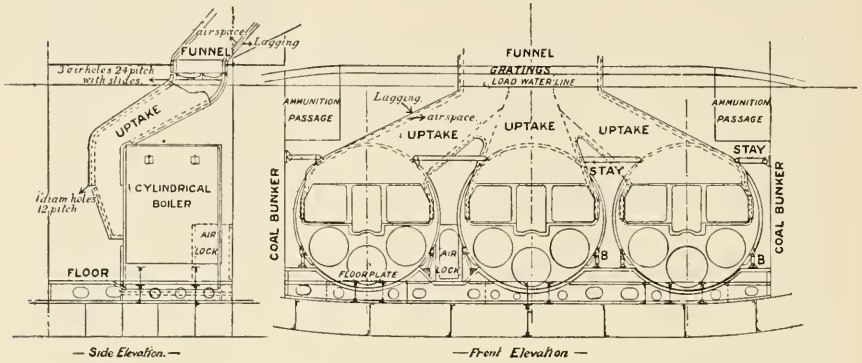


Fig. 81.—Cylindrical Boiler Uptakes.

it. The air from the stokehold has free access to the space between the two casings, and is allowed to escape either into the stokehold at the highest point, or into a space formed by a continuation of the second casing to some height above the upper deck and around the funnel.

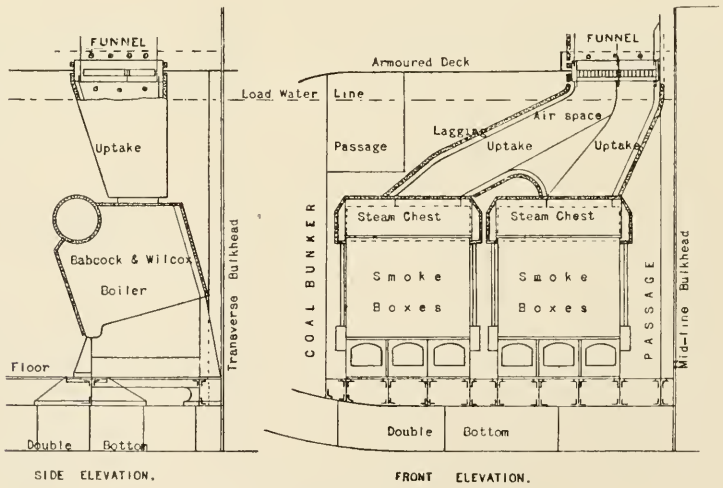


Fig. 82.—Water-tube Boiler Uptakes.

The smoke-box doors are also fitted with a second casing in the same way, but in this instance the hot air escapes immediately above the doors. Another thin system of plating (generally galvanised

plates are used for all casings) is fitted across between the boilers and parallel to the front of them at about 18 to 24 inches back from the front end. This casing prevents the dust and ashes being carried into the space behind, and, with closed-stokehold forced draught, assists in maintaining the air pressure near the furnace fronts. Doors for convenience of access to the parts behind are fitted as necessary. In some cases, the space so formed on each side of the boiler is utilised to warm the air which may be supplied to the fire near the back end of the grate. It may be seen in the illustrations of Yarrow, and Thornycroft boilers, and is also used with other types.

Air-Supply and Brickwork.—In some water-tube types air is admitted at the back of the fire as well as at the front in the usual way. The boilers, generally used for high rates of forcing and combustion, are fitted with self-closing doors on the ashpits, both at back and front. Arrangements are made (see Yarrow-Howden system, Fig. 86) for keeping the doors at the back closed when required. The casings are generally made double, and the space so formed is utilised for heating some of the air supplied for combustion. In a few cases this warm air is supplied below as well as above the grate. The circulation of air through these spaces keeps the casings cool, and thus, in addition to promoting efficiency of combustion, it keeps the stokeholds cool. The air doors in the casing are shown in the various figures. The bursting of a tube, although merely a temporary disablement of a boiler, is sudden, and the necessity for automatically self-closing doors to the fires, ashpits, and air-supply is apparent.

Brickwork is necessary, to a more or less extent, in all water-tube boilers, and is employed principally to protect the casings surrounding those parts of the furnace which are not flanked by the tube heating surfaces. The weight of brickwork is considerable, and in some early cases exceeded that of the actual boiler without water. Recently a very light and durable brick, or fire tile, has been introduced, and is used for those parts (and for baffles) which are not subject to blows from the firing tools. The bricks or tiles are sometimes secured in place by inset bolts to the inner galvanised steel casing; but the brickwork should, however, be as far as possible self-supporting, and should never be entirely dependent on the bolted attachment. Many engineers in oil-fired vessels place old bits of glass, broken bottles, etc., in the furnaces, and this fuses and forms a good protective glazing on the brickwork which is beneficial and adds to durability. Various systems are shown in the figures, and in some boilers it should be

noticed that conduits are formed in the brickwork for the air-supply above the fire.

Funnel.—The smoke and hot gases pass into the smoke boxes from the boiler, and are conveyed through passages, called the uptakes, to the funnel, as shown in Figs. 81 and 82. The area of opening of the funnel is usually from one-seventh to one-tenth the area of the firegrate. The plates separating the several uptakes are carried a sufficient height up the funnel, generally as high as the armour deck gratings, to cause the gases to move in approximately the same direction when they mingle together, and thus avoid loss of draught from shock or confusion of currents.

The funnel is surrounded (see Fig. 84), generally throughout its entire length, by a similarly shaped structure, called the outer funnel; holes are provided at the bottom to admit air into the space provided between the two, and at the top a bonnet extends outwards from the top of the inner funnel and overlaps the outer, leaving a passage-way for the heated air, and preventing rain from obtaining access to the space between them. The two funnels are stayed together by long bolts or rivets with distance pieces.

An additional casing is fitted at the lower part of the funnel, forming another air space. The heat radiated from the funnel is thus again intercepted, and the ascending current of air is carried up into the atmosphere a sufficient height above the upper deck to prevent inconvenience to the people on board. This outer casing is also used as the outlet for ventilating the bunkers and some other parts of the ship.

Telescopic Funnel.—In ships fitted with sail as well as steam power, the funnel was sometimes made to lower when not required. The upper part, nearly one-half, could be lowered inside the lower part, or hoisted again by means of chains and pulleys, worked either through worm or pinion gearing.

Funnel Guys.—The funnels are generally long compared with their bases, and have a tendency to topple over when in a sea-way. To prevent this wire-rope stays are attached to a band fitted round them at about two-thirds of their height, and secured through right- and left-handed stretching screws to eyebolts at the ship's side, or other convenient part. These stays are sometimes eight in number, and are radiated from the funnel to give support in all directions. The funnel lengthens when heated, and before lighting fires the guys should be slacked off a sufficient amount to allow for this extension.

Very long funnels have an additional band round the funnel, about 6 feet from the top, and a second set of guys is attached to it.

Funnel Covers. — To prevent the deposit of moisture on the interior of the funnel when not in use, covers are fitted above them. They should overlap a moderate amount, and at the same time sufficient space should be allowed for the egress of the smoke from an auxiliary fire or airing stoves. Thin sheet-iron or steel plates were frequently used ; but canvas covers are now generally supplied, and are more convenient for shipping and unshipping. The cover is supported by crossed rods raised over the mouth of the funnel ; the canvas is spread over them, and laced down to the edge of the funnel top. When using airing stoves a corner of the cover can be turned back. The covers are put in place from the interior of the funnel, and a platform is fitted for this and other purposes about 4 feet from the top, and access is obtained by iron ladders permanently fitted inside and leading up from the armour gratings. Doors are fitted in the several casings for access to the interior of the funnel and uptakes.

PART III

COMBUSTION

CHAPTER X

STOKING AND PRACTICAL CONSIDERATIONS

Introductory.—Combustion is the chemical combination of a combustible with oxygen at a sufficiently high temperature.

The oxygen is supplied by the surrounding atmosphere, and in certain cases, now becoming very usual, artificial means are employed to increase the supply. This process is adopted in most steamships, and is called forced draught.

Average samples of fuel vary considerably; they are approximately composed of the following chemical substances:

	Carbon, per cent.	Hydrogen, per cent.	Oxygen, per cent.	Sulphur and Ash, per cent.	Heat, B.T.U.	Actual Evaporation, from and at 212°, in lb. of water.
Scotch coal	78·5	5·6	9·7	6·2	14,100	9·5
Newcastle (N.S.W.) coal .	82·0	4·5	4·0	9·5	14,350	9·26
Newcastle (U.K.) coal .	82·0	5·0	6·0	7·0	14,820	9·83
Welsh coal	84·0	5·0	4·0	7·0	14,858	10·17
Westport (N.Z.) coal . .	84·0	7·8	4·0	4·2	14,800	10·97
Patent fuel	84·0	5·0	3·0	8·0	15,000	11·00
Petroleum, crude Beaumont	84·6	10·9	2·87	1·63	19,081 (by calorimeter)	

Of these substances carbon and hydrogen form the combustibles, and by their union, chemically, at a high temperature with the oxygen of the atmosphere combustion is effected. It is true that combustible substances will unite with oxygen at all temperatures; but such

union is effected so slowly that, for all purposes for which the fuel is required, it may be considered as practically non-existent.

Spontaneous Ignition. — The chemical combination of combustibles with oxygen at low temperatures must not be ignored. Coals exposed to the action of the open air, sun, and rain become deteriorated by slow combustion. If totally immersed, there is some reason to believe that the calorific value may become greater; for some New Zealand coal gives out a greater quantity of heat than the best Welsh, notwithstanding a more than moderate amount of wetness. For slow combustion to become active, it is necessary that the air should have slight access to the fuel, and be almost stagnant. A small quantity of heat is then generated by slow combustion, and the consequent rise of temperature fosters the same process between other particles; and the oxygen contained in any dampness present assists the process. These combinations proceed with increasing rapidity as the temperature increases, until some disturbance takes place, admitting more air, when the carbon gases evolved burst into flame, in some cases explosively.

Furnace Combustion.—It is only at a high temperature that a common fire burns, and if from any cause the temperature of the surrounding substances be lowered, the fire will cease to burn briskly. When coal is heated gases are given off, represented by the chemical symbols—

CH_4 = light carburetted hydrogen or marsh gas, fire-damp.
C_2H_4 = heavy ,, ,, olefiant gas.
C_2H_2 = acetylene
C_6H_6 = benzine
$C_{10}H_8$ = naphthaline
} and others of similar composition.

The first two are the most important; and, as far as the furnace of a boiler is concerned, the others may generally be ignored.

In a boiler furnace, under ordinary conditions, there are various distinct stages of combustion.

1. If Welsh coal be heated (bituminous coal is used in the ordinary production of gas for lighting purposes), over 20,000 cubic feet of hydrocarbon gases are distilled off from each ton of coal so treated; this leaves a solid residue, which is practically all carbon. Thus, when coal is thrown on the fire in the furnace heat is first used to distil off these gases. At a sufficiently high temperature hydrogen gas is also given off.

2. If the temperature be sufficiently great these gases first separate into carbon and hydrogen, then chemically combine at a temperature of about 1400° Fahr. with the incoming oxygen of the air, and form oxides of carbon and hydrogen; combustion ensues, producing **heat.** All the air necessary for this stage, if supplied at a sufficiently high temperature, might be injected above the fire—that is, through the furnace door, or thereabouts.

3. The air supplied below the fire is at first warmed by the fire, then the oxygen in it combines with particles of solid carbon on its way through the fire, generally forming carbon monoxide and dioxide (CO and CO_2); and if the temperature be high, together with a sufficiency of oxygen, the CO is converted into CO_2 before it leaves the furnace.

The **funnel gases** are generally carbon dioxide, or CO_2; steam gas, or H_2O; and nitrogen, or N, which forms about 77 per cent by weight of the air supplied for combustion.

Rate of Combustion.—The weight of fuel burnt per square foot of grate surface per hour is called the rate of combustion.

In the Merchant Service, when working economically with funnel draught only, the rate is from 15 to 20 lb. of coal. It is now more usual to adopt some form of forced draught—Howden's closed ashpits is most frequent—or of induced draught, when the rate is increased to 24 to 40 lb.; but at the same time the length of grate is frequently decreased, so that the power per unit of boiler is not increased to quite the same extent. The improved systems of forcing have during the last few years, however, increased the power obtained from similar boilers by 40 to 60 per cent, and in one or two instances by as much as 90 per cent.

Lighting Fires.—When lighting fires in a boiler of the cylindrical type, with more than one furnace, the lower fire or fires should be lighted up before the others; generally two or three hours are allowed to warm the boiler gradually by promoting sufficient convection current for the purpose. The fires inside the several doors of a water-tube boiler should all be lighted at the same time.

The furnace is *primed* by throwing on coal to an even thickness of about 3 or 4 inches. Just inside the furnace door wood and coal are topped up ready for lighting, and should be so arranged that all air entering the furnace door must also pass through the topping. A piece of oily waste is generally used for lighting; the furnace door is left partly open and the ashpit door closed until the topping is well

alight and ready for spreading and pushing back, which is done with a rake. After spreading the fire the furnace door is closed and the ashpit door partly opened, and the draught regulated by the amount of opening. Fresh coal is added as necessary, and the fire gradually levelled up to a thickness of about 6 inches.

Funnel Draught.—Before lighting fires, a draught or current is created by the air in the boiler room being somewhat hotter, and therefore lighter per gallon, than the air at the mouth of the funnel. The lighting of the fires creates a still stronger current, because the temperature inside the funnel increases. The current is also increased by greater height of the funnel. It is wasteful to allow the funnel temperature to become too high, because heat is then passed away into the atmosphere and no useful work obtained from it. Thus when this temperature is moderate, between 600° and 700° F., the draught is practically dependent on the height of the funnel above the firegrate, and if no artificial means are employed to assist the draught it is termed *Natural draught*.

The general arrangement of four Babcock and Wilcox boilers is shown in Fig. 83 of the Ice-Breaker *Montcalm*. Air is supplied to the stokeholds and fires through circular trunks leading from cowls above decks to within about 4 feet of the stoking platforms. These trunks are also used for getting up ashes. In the left-hand figure, showing the longitudinal view, the safety valve lifting gear may be seen, also the feed check regulating gear in both views; these arrangements are workable from the stokehold platforms. Below the boilers is a tank built as part of the ship which may be used for water storage. The tank top forms in many mercantile vessels an inner bottom.

In destroyers and other small vessels only a very little height of funnel can be fitted, and some system for increasing the draught —called *Forced draught*, but actually assisted draught—is used.

Forced Draught.—The primary object of forced draught is to burn more coal per square foot of grate surface, or oil fuel per cubic foot of combustion space, and thus obtain a greater power from a given weight, and also on a given floor area, without increasing the funnel temperature and consequent loss of heat.

The principal methods in ordinary practice are:

1. Closed stokeholds, when the only outlet for the air under compression is through the fire. (Plenum.)

2. Closed ashpits, when the stokeholds are open to the atmospheric

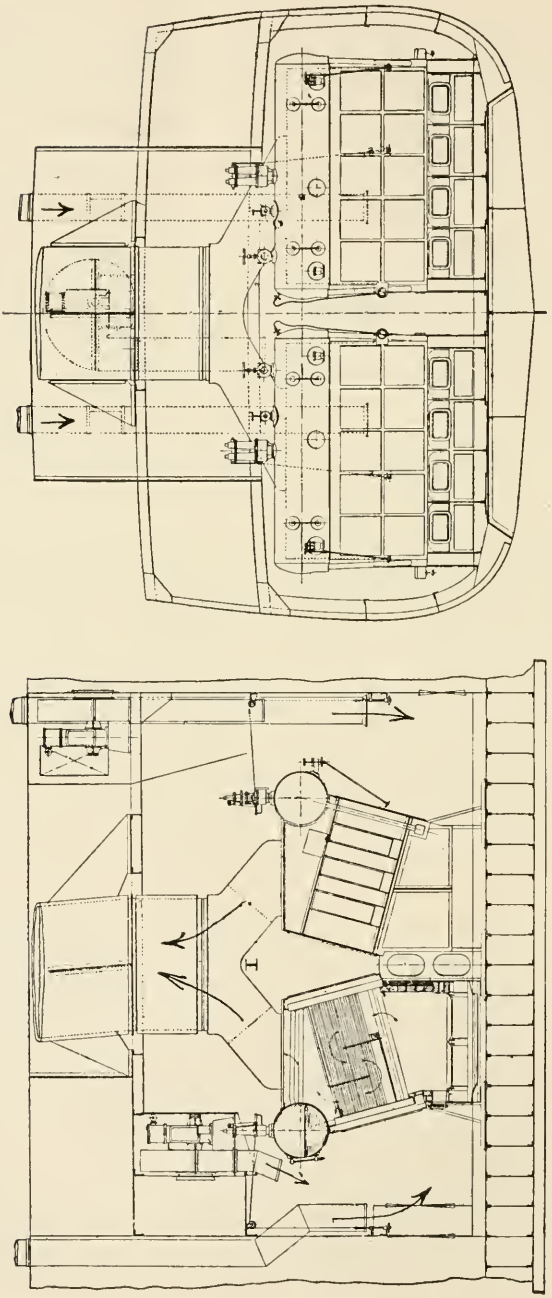


FIG. 83.—General Arrangement, Babcock and Wilcox Boilers, Ice-Breaker *Montcalm*.

pressure, but the furnaces are supplied with air, compressed, and generally heated previous to supply. (Plenum.)

3. Suction draught, by drawing up the products of combustion from the furnaces through the usual passages. (Vacuum.)

4. Steam or exhaust blast, inducing and accelerating the draught in the funnel. (Vacuum.)

5. Jets of compressed air blown into the base of the funnel and inducing a draught, now practically abandoned as too cumbersome. (Vacuum.)

6. Jets of liquid fuel blown over the incandescent coal. This method increases the rate of combustion, but the air is not necessarily supplied artificially.

1. **Closed Stokeholds.**—This system has been frequently adopted for increasing the rate of combustion in warships. It is a ready method of increasing the power obtainable, but it is the *least* efficient means of burning coal fuel. For warships, where the full power, however, is seldom required, it answers the purpose for which it is fitted; but in the mercantile marine, where the primary object is to obtain an economy of fuel, it is very properly excluded. The extra weight is less than that required with either suction, closed-ashpit, or the compressed-air systems. Water-tight compartments already exist in warships, and the only additions necessary are a few thin screen doors of sufficient strength to resist the very light air-pressure of the excess, or plenum, over that outside the screens. Air locks, as shown in Fig. 84, for twelve Yarrow boilers, with double doors are usually fitted for entry to and exit from the stokehold, so that no sudden decrease of pressure should produce a back draught from the furnaces. An objectionable feature of the closed stokehold is that the men on watch are shut in, and outside supervision is very difficult.

A necessary adjunct of the closed stokehold is that the air-supply must be cold, and the admission of considerable quantities of cold air when the fires are cleaned. The admission of cold air has been disastrous to cylindrical boilers when forcing above a rate of combustion of about 22 lb. per square foot of grate, especially in boilers of the common combustion-chamber type. Such failures are caused by lack of free circulation in the boiler and lack of proper allowance for difference of expansion in the various parts.

2. **Closed Ashpits.**—Apart from the supply of hot air to the furnaces for combustion, this system has some advantage over others, and in combination with such heating it is the most economical.

Howden's system of closed ashpits and hot-air supply has been extensively tried and proved. It is fitted in a large number of ocean-going steamers. In almost every case there has been a gain in economy of fuel, and in many a saving in space has also been effected.

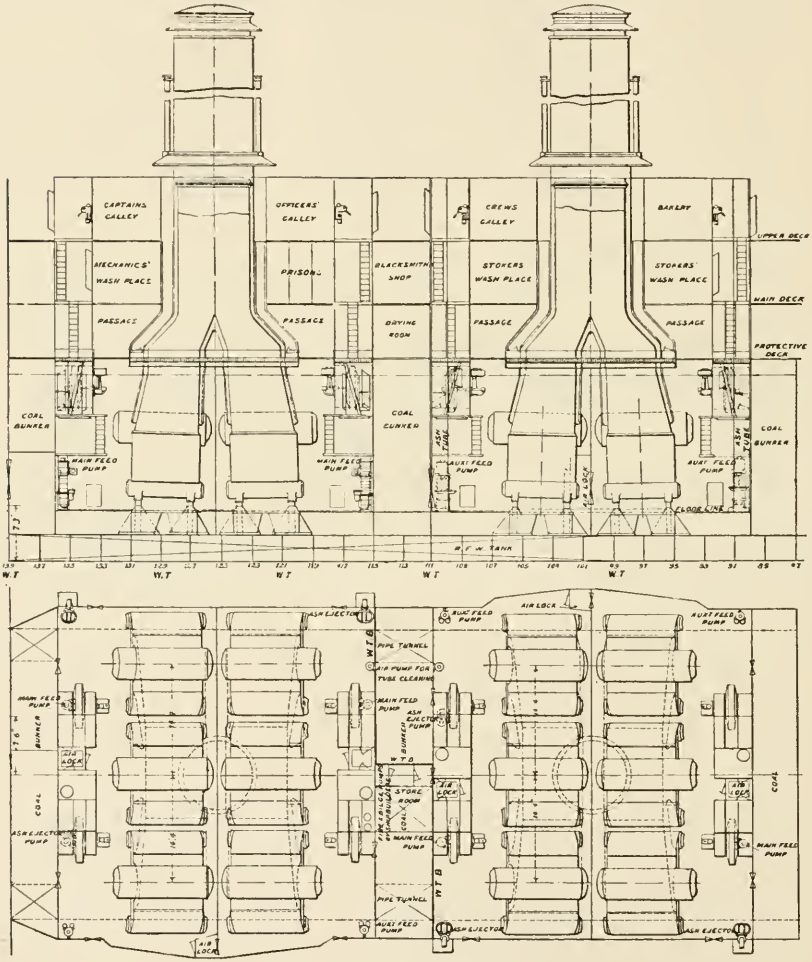


Fig. 84.—Closed Stokeholds.

The hot air-supply system consists essentially in a number of very thin tubes fitted vertically in the smoke box just above the exit of the gases from the boiler heating tubes, as shown in Fig. 85. The gases pass through these tubes, internally, on their way to the funnel, and

give up a large amount of their heat to them, which in turn pass it on to the air-supply from their external surfaces. The air-supply to the boiler-room comes down the usual ventilators (either with or without cowls on top), and is forced by a fan, through a trunk delivery, among the tubes, externally; thence downward through passages provided around the casing of the smoke box and in front of the boiler, and

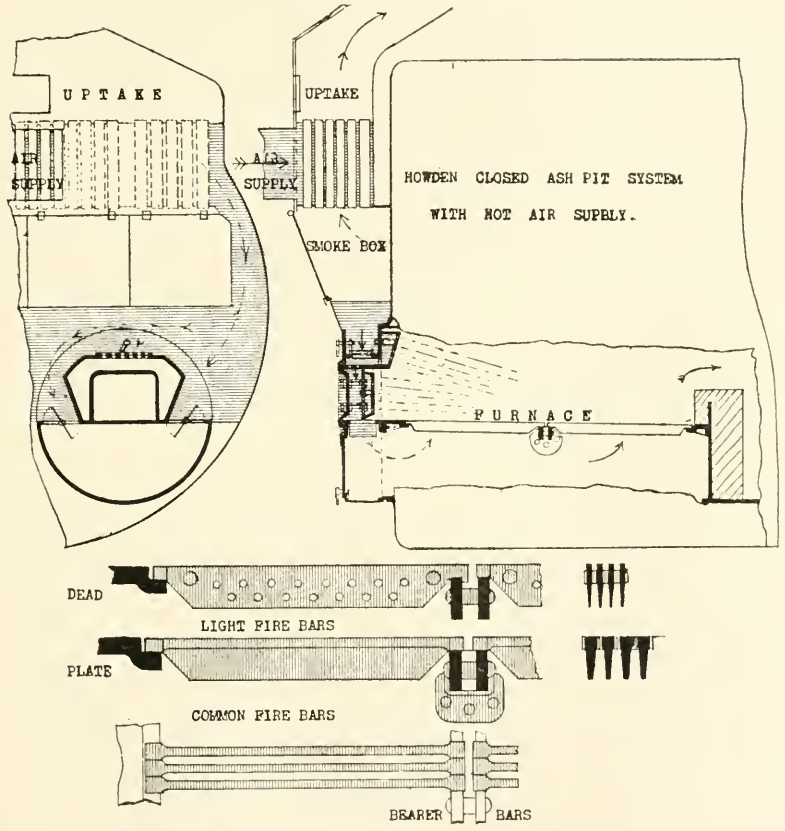


FIG. 85.—Closed Ashpits.

through holes in the furnace front both above and below the firegrate, in the proportion of about 1 to 3. By a simple system of levers it can be arranged that, when either the furnace or the ashpit door is opened, the draught can be automatically and independently shut off—thus a possibility of accident is easily prevented; but this fitting is usually unnecessary. If the boilers are suited to high

rates of combustion, the system can be fitted conveniently to water-tube boilers.

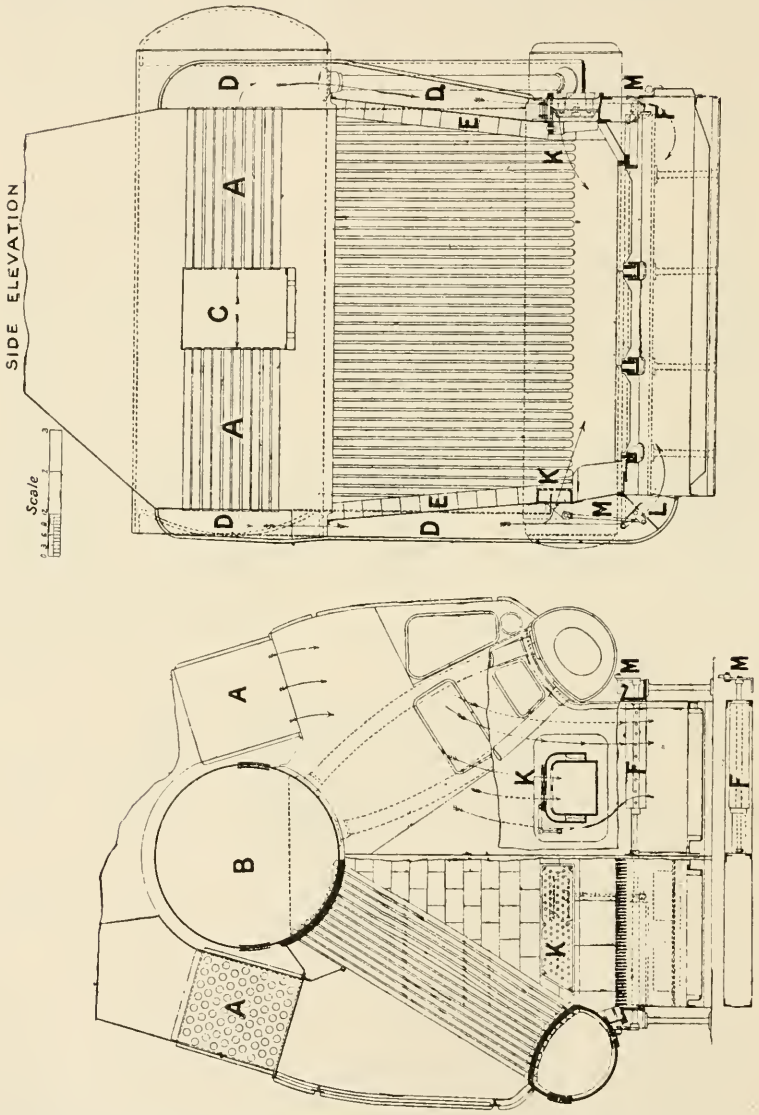


Fig. 86.—Closed Ashpits and Hot-air Draught.

Howden's system of closed ashpits and hot air-supply has been fitted in conjunction with Yarrow boilers, one of which is shown in Fig. 86. There are four nests, *A*, of air-heating tubes fitted in the

uptakes, two on each side of the steam drum, *B*. Air is drawn through the boiler-room and ventilators, and pumped by centrifugal fans through trunks into the space *C*; the air then passes through the tubes, *A*, into the front and back end casings, *D*, of the boiler, between the brick-work casing, *E*, and the outer casing, from which it supplies the fire above the grate (at *KK*), and below (at *L* and *F*) through air-doors and slides. By a simple system of levers, *M*, connected with the hinges of the fire-doors, the air-supply is automatically shut off while a fire-door is open. The ashpit doors are always closed except when drawing ashes, etc., and the pressure in the ashpit may be (equal to 2 or 3 inches of water in some cases) in excess of the pressure (atmospheric) in the stokehold. The temperature of the air-supply to the furnace is from 180° at ordinary powers up to 300° or more at full power, and naturally a high economy of fuel should result because the heat supplied to the air is taken from the funnel gases. The air-heating surface is about thirteen times the grate area, and the tubes $2\frac{1}{2}$ inches diameter.

Great care should be taken in fitting up the lead of pipes from the ventilating trunks to the furnaces, in order that the loss of head, due to friction in the pipes, bends, etc., shall be as small as possible; otherwise the increased power required to drive the fan increases the expenditure of fuel unnecessarily, and the fan itself becomes cumbersome, and very liable to break down. The lead should be fair, with as little abrupt alteration of direction of current as possible.

The **supply of hot air** to the furnace tends to reduce the quantity practically necessary for combustion; probably 18 to 20 lb. of air per 1 lb. of coal are sufficient. This reduces the velocity of the furnace gases on their way to the funnel, and, allowing them more time in which to impart their heat to the boiler, consequently reduces the funnel loss. It may be taken that, approximately, any reduction in the funnel temperature which is employed in heating the furnace supply is a double gain, and thus the efficiency of combustion is greatly improved: the furnace temperature is higher, greater heat can be imparted to the water, and combustion is more complete. The reduction of funnel temperature and the casing in of the smoke boxes by the screen plates for conducting the downward flow of the warmed air, the temperature of which is considerably lower than that of the smoke box, conduce to a much cooler ship and stokehold, which is much appreciated in hot climates.

3. **Suction Draught.**—The great advantage of this system—in

fact, any system which does not involve entirely shutting off the stokehold—is that the stokeholds are always open, comparatively clean, cool, and comfortable, and the ventilation of these parts is better. The situation of the suction fans in the base of the funnel increases the wear and tear, and the volume of the hot gases passing through the fan is nearly double that in other systems with cold air-supply. This might be objectionable ; but if the air is heated to nearly the same temperature as that of the escaping gases, then it appears that the volume of gases is practically the same in any system. Possibly only about 10 per cent more power may be required to drive the larger fans for the suction system as opposed to, say, Howden's hot-air and closed ashpit system.

4. **Steam** *or* **Exhaust Blast.**—The steam blast is a ready method of inducing a draught and thus increasing the rate of combustion. Its use has been generally abandoned for marine purposes because it is so wasteful both in fuel and water.

The exhaust blast, as used in locomotives, is probably the most prolific method of increasing the rate of combustion. The highest known amount, viz. from 200 to 224 lb. per square foot of grate, is seldom attained, but 120 lb. is common. This rate does not produce any immediate defect in the boiler, but it no doubt shortens its life. The exemption from immediate defects is entirely due to the mode and manner of working and forcing. The draught produced by the exhaust is exactly proportional to the power taken from the boiler. The damage which generally follows from the admission of cold air is also reduced to a minimum, because the stoking is almost entirely done when the engine is stopped, or at low speeds and powers. No fire-cleaning or other operation which necessitates the prolonged opening of the fire-door is necessary while the engine is running at full power and with the blast full open, for this last condition is generally required only for short periods ; this practice is entirely impossible in sea-going ships. The water required to make up that lost up the funnel is very great, probably over 70 per cent of the total water evaporated, because only a small amount is deposited in the exhaust tank. The fires must be stoked to maintain a continuous supply of steam for full power, which in marine practice entails cleaning them at frequent intervals, when cold air must also be admitted, resulting in serious damage to the boiler.

Air-Supply.—The supply of air below the fire must be properly regulated. The fire bars must be spaced a proper distance apart,

according to the type of fuel used. For the best Welsh coals, which are quick-burning, the area of opening between the bars for moderate rates of combustion is generally about 50 to 55 per cent of the total fire area. With North-country and N.S.W. coals, which are relatively slow-burning, the proportion of spacing should be decreased to about 35 or 45 per cent. For forced draught with Welsh coal the proportion of opening is increased to 60 per cent.

To allow the air sufficient passage through the fire, the average thickness of the fire should never exceed 6 inches for quick-burning coals, and about 8 inches for slow-burning coals.

If the air-supply to the furnace is heated before it reaches the fire, it tends to reduce the formation of smoke, because the consequent increase of furnace temperature produces more complete combustion.

Stoking.—Economy of fuel can be obtained by careful and regular stoking, and the coal should be handled with a shovel, care being taken to have the coal small enough to part with its gases easily, and at the same time not small enough to fall through into the ashpit. The coal should be thrown on evenly, about four or five shovelfuls at a time, and the average thickness of the fire should not be greater than 6 inches. With a fire of 5 inches thickness the supply of air should be sufficient to check smoking, and at the same time give sufficient air for complete combustion of the fuel.

Under forced draught a *thin* fire is also necessary for good combustion. The air pressure, or *plenum*, can be considerably reduced by efficient stoking, and for this reason the amount of air pressure is a very unreliable measure of the rate of combustion, unless the fires are well stoked and are kept a proper thickness. Good stoking alone will increase the indicated horse-power or evaporation of a boiler by at least 10 per cent, and instances have been observed where the increase has been over 25 per cent.

The fires should be levelled with a rake about every half-hour, and *holes in the fire* should be guarded against, because a large volume of cold air may be admitted in this way. If the gases go up the funnel as carbonic oxide, only 4400 thermal units are given out; but if it can be properly burnt by a sufficient supply of oxygen, 14,500 units can be imparted to the boiler for each 1 lb. of coal; therefore, within ordinary limits, it is better to have too much rather than too little air supplied. Because the coal has disappeared from the grate it does not necessarily mean that it has been usefully burnt; both carbonic oxide and carbon dioxide are invisible, and thick fires, through which a

sufficient supply of air cannot penetrate to the part above the fire for combustion, should be carefully avoided.

The fire bars should be cleared by a pricker from below the grate, not from above, because this entails opening the fire-door and admitting cold air to the furnace, with an attendant lowering of the furnace temperature. The furnace doors should be kept closed, and opened for as short a time as possible when firing. When cleaning the fire the ashpit damper should be kept closed; in short, the furnace door and the ashpit damper should never be open at the same time. A routine of firing should be laid down suitable to the boilers, and always carried out, so that the green coal does not entirely cover the grate.

An accumulation of ashes is a thing to be guarded against, and they should be drawn out at least once in each four-hour watch; they should then be sifted or riddled, and the small coal thrown back on the fire. Ashes settle in or among the tubes, and these must be removed at regular intervals; and, as opportunity offers, the heating surfaces should be cleaned. Soot is a very bad conductor of heat, and consequently any slight scale on the heating surface considerably reduces the evaporative power. For cleaning the interiors of the tubes in cylindrical boilers, forked scrapers or button-ended scrapers are used, and steel-wire brushes are at all times most useful. The removal of *retarders*, which now form part of cylindrical boiler plant, also materially assists in keeping the tube internal surfaces clean. After removal, these should be cleaned with a wire scrubber before they are replaced.

When *banking fires*, they should be banked in front if banked at all; generally this is unnecessary. The fires should be allowed to burn down as low as possible, and a little fire kept alight at the front end, in preparation for pushing back when required. If the fires are dirty, the clinker should be taken out after breaking it up by a pricker applied below the grate; but the removal should not be carried out until the fire is burnt down very low, to reduce the possible damage from the admission of cold air. Care should be taken that the fire is kept well back from the dead plate at the furnace door. The intelligent carrying out of these details tends to the reduction of strains on the boiler and freedom from leakage, especially at the furnace and tube-end joints.

Smoke.– Smoke is the *visible* result of incomplete combustion of the tarry constituents of the fuel, and in practice is caused by the temperature of combustion being below that of the ignition-point of these tarry constituents. Of the three gaseous products—steam,

carbon dioxide, and carbon monoxide—of combustion, steam alone plays an important part in the formation of smoke, whilst the other less important constituents are tar vapour, minute particles of unburnt carbon, and ash, drawn upwards by the funnel draught. The amount of carbon in smoke is far less than is generally supposed, even when given off by bituminous coal under the imperfect conditions of combustion existing in an ordinary fire-place. (See *Cantor Lectures*, 1906, Professor V. B. Lewes.)

Smoke consists of vesicles, globules, or bubbles of nitrogen, carbon dioxide, carbon monoxide, traces of oxygen and such hydrocarbons as methane, enclosed in skins of condensed vapour, and liquids from the burning substances which give rise to them. If collected, and the tiny vessels (which are very light, and float easily in the atmosphere) are burst by friction, a liquid " tar " is obtained and the gases escape. In the production of ordinary lighting gas the tarry liquids are removed by distillation at a comparatively high temperature, and the coke remaining, and consisting of almost pure carbon, burns smokelessly, but only ignites and continues to burn at a high temperature. But if the temperature of distillation be comparatively low and only just sufficient to expel the tarry constituents, principally in the liquid state, an intermediate form of coal and coke, now known as " Coalite," remains, which ignites at a comparatively low temperature and is stated to produce no smoke.

Soot is produced when a strong draught carries up the gaseous bubbles and particles of carbon dust. Friction against the sides of the flue and chimney passages bursts the bubbles, and the liquids of which the skins are formed adhere to the surfaces in conjunction with carbon dust. The liquids are generally vaporised or burnt away, leaving almost pure carbon dust in the form of soot in various quiet places formed by the flue passages.

In practice the *prevention of smoke* formation from coal and oil fuel is attainable only by careful stoking, ensuring good combustion at a temperature high enough to completely burn the tarry matters contained in the fuel. Boilers which are most likely to produce smoke are those in which the space for combustion is unduly contracted, and in which the furnace height is small, and where, as in the Belleville, Dürr, Niclausse, Babcock and Wilcox, and other boilers generally of the horizontal water-tube type, the cold surfaces of the generating tubes are in close proximity to the fire. It is quite possible to produce excessive smoke by supplying too much air, and thus reducing the

furnace temperature by this means below that necessary for proper combustion.

Flaming.—An unnecessarily thick fire does not allow sufficient air to pass through it to complete the combustion of the gases given off by the fuel. Carbon monoxide, CO, is thus produced, and this *invisible* gas at a high temperature bursts into flame when it comes into contact with more air, including oxygen, at the mouth of the funnel; the CO burns into CO_2, carbon dioxide, and produces the flames seen. Flaming can be almost instantly stopped by slightly opening the furnace door, thus increasing the air-supply above the fire; and an opportunity should at once be made to level off the fires, to clean them if necessary, and to reduce them to an even thickness. Flaming is most likely to occur with water-tube boilers, but is most noticeable in small vessels with short funnels. It is seldom noticed with cylindrical boilers because the gases have a longer and more circuitous route to travel, which tends to reduce the temperature at the funnel mouth below that at which the carbon monoxide will take fire. Accidental flaming may occur for an instant when under a strong forced draught; but, as pointed out above, it is always preventible, it can be instantly checked when known, and it is entirely dependent on the stoker.

Periodical Cleaning of Fires.—In practice the fire requires cleaning after about 2 cwts. of coal have been burnt on each square foot of grate surface; the time and the amount depend on the rate of combustion, and also the class of coal used. The heating surfaces, on the side next the fire and furnace gases, require cleaning after about 6 cwts. have been burnt per square foot of grate. The necessity for cleaning the fire limits the time for which the full power of all the boilers can be maintained. Thus, when burning 60 lb. or over per square foot of grate the full power is limited to about 3 or 4 hours; when burning 30 lb. the full power is limited to about 8 hours—or, when burning 15 lb., to 16 hours—without cleaning fires. But, of course, by a sufficient reserve of boiler power the full *engine* power can be obtained for any period. Generally the boiler power of merchant vessels is sufficient to maintain the full engine power, and the reduction of sea-going speed is generally inappreciable.

CHAPTER XI

EFFICIENCY AND THEORETICAL CONSIDERATIONS

Calorific Value.—This is the quantity of heat which is given out by 1 lb. of the fuel when combustion is complete. It is measured in British thermal units, and for engineering purposes the value may be calculated either on the *higher*, or the *lower* scale.

The *higher* calorific value is the total heat given out during complete combustion, and is generally based on the value determined by experiment when using a calorimeter.

The *lower* calorific value assumes that, in practice, some of the heat generated by combustion cannot be utilised, particularly that portion which is carried away by the hydrogen, and escapes with the products of combustion as steam, either in a saturated or superheated state. In all calculations relating to the thermal efficiencies of engineering processes the lower scale is used as the standard of comparison.

The higher calorific value of hydrogen is about 62,032 b.t. units. During combustion 1 lb. of hydrogen combines with 8 lb. of oxygen to form 9 lb. of steam, and when mixed with other products of combustion, each 1 lb. of this steam carries away at least 966 b.t. units (the latent heat of 1 lb. of steam at atmospheric pressure). The lower calorific value of hydrogen, as applied to steam-raising, or to internal combustion engines, thus becomes

$$= \text{Higher calorific value} - 9 \times 966$$
$$= 62{,}032 - 8694$$
$$= 53{,}338 \text{ b.t. units per 1 lb. of hydrogen.}$$

The quantity of heat generated by the combustion of 1 lb. of *carbon* is found by experiment to be about 4400 b.t. units if converted into CO, or carbon monoxide ; or about 14,500 b.t. units if converted into CO_2, or carbon dioxide ; and if the supply of air is sufficient, and the

temperature suitable, it may be concluded that all the carbon is so converted.

In the usual formula for calculating the calorific value of fuels, the fractions of carbon, hydrogen, and oxygen contained in 1 lb. of the fuel are represented by their chemical symbols, C, H, and O. Dulong, to whom this formula is attributed, generally used the higher calorific value, but the lower value is now generally used, and is more in accordance with engineering practice.

In any fuel the hydrogen available for combustion is $\left(H - \frac{O}{8}\right)$, because $\frac{O}{8}$ lb. of the hydrogen is already combined. Thus the lower calorific value, in b.t. units per 1 lb.

$$= 14{,}500\left\{C + 3{\cdot}68\left(H - \frac{O}{8}\right)\right\}.$$

The *influence of temperature* on the products resulting from combustion is of great importance. From experiments by Dowson and Larter, it was found that carbon dioxide and hydrogen are produced at combustion temperatures below 1112° F.; carbon dioxide and monoxide, and hydrogen between 1112° and 1832° F., the proportion of monoxide and dioxide being dominated entirely by the temperature; and above 1832° carbon monoxide and hydrogen are produced.

In a boiler furnace such interactions also take place, but eventually owing to the ample supply of air it may be assumed that the fuel is converted into (and escapes up the funnel as) carbon dioxide and water-vapour. The final temperature of the products of combustion where they leave the boiler heating surfaces should be therefore below 1112° F., and above this temperature it may be concluded that combustion is incomplete, and that there is a consequent loss of economy.

Evaporative Unit.—For the comparison of the calorific value of various fuels used for steam-raising, it is convenient to have a common basis, and this is the weight of water which 1 lb. of the fuel, assuming the whole of the lower calorific value to be utilised, can convert from a temperature of 212° F. into steam at the same temperature and under normal atmospheric pressure. For such a conversion only the latent heat (966 b.t. units per 1 lb. of water) is required, and the variable quantities of total heat and sensible heat are omitted. The evaporative unit is thus equal to the lower calorific value divided by 966; that is—

$$= \tfrac{14500}{966} \left\{ C + 3{\cdot}68\left(H - \frac{O}{8} \right) \right\}$$

$$= 15\left\{ C + 3{\cdot}68\left(H - \frac{O}{8} \right) \right\}$$ in lb. of water evaporated *from* and *at* 212° F., per 1 lb. of fuel.

Applying this to the proportions for average Welsh coal, the evaporative unit—

$$= 15\left\{84 + 3{\cdot}68(5 - \tfrac{1}{2})\right\} \div 100$$
$$= \tfrac{15}{100}\ (84 + 4{\cdot}5 \times 3{\cdot}68)$$
$$= 15 \text{ lb. about, for each lb. of Welsh coal.}$$

In practice this is seldom obtained; a fair average result is only about 9 lb., but under good conditions as much as 12 lb. of water are evaporated per 1 lb. of good Welsh coal. For Admiralty tests *on shore*, the amounts required are—

11 lb., when burning not less than 30 lb. per sq. ft. of grate.

$11\frac{1}{2}$	,,	,,	,,	24	,,	,,	,,
12	,,	,,	,.	18	,,	,,	,,

The respective evaporative efficiencies being 73·3, 76·6 and 80 per cent.

Air-Supply *by weight.*—Oxygen is found mechanically mixed with nitrogen in atmospheric air in the proportion by weight of about 23 to 77, so that for the supply of 1 lb. of oxygen 4·35 lb. of air are required. For complete combustion, 1 lb. of carbon requires 2·66 lb. of oxygen, or—

$$4{\cdot}35 \times 2{\cdot}66 \text{ lb.} = 11{\cdot}6 \text{ lb. of air.}$$

For complete combustion 1 lb. of hydrogen requires 8 lb. of oxygen, or—

$$4{\cdot}35 \times 8 \text{ lb.} = 34{\cdot}8 \text{ lb. of air.}$$

But part of this hydrogen is already contained in the fuel itself, and only $\left(H - \frac{O}{8} \right)$ lb. require air supplied, so that—

Total weight of air required for complete combustion of 1 lb. of fuel $= 11{\cdot}6C + 34{\cdot}8\left(H - \frac{O}{8} \right)$

$$= 11{\cdot}6\left\{ C + 3\left(H - \frac{O}{8} \right) \right\};\ \text{or}$$

$= 11{\cdot}3$ lb. for about 1 lb. of Welsh coal.

In practice this amount is insufficient, and from 15 to 24 lb. must be supplied respectively for a very efficient artificial draught, and for natural draught, due to height of funnel only.

The *volume* occupied by 1 lb. weight of air at ordinary atmospheric pressure and at a temperature of 62° Fahr. is about 13·14 cubic feet, and therefore about 177·1 cubic feet of air must be supplied for each 1 lb. of coal burnt under artificial draught, and 315·4 cubic feet for funnel draught. And for *each ton* of coal burnt under the above conditions, 396,704 and 706,406 cubic feet are required respectively.

These volumes of air must be considered when calculating the passage areas of ventilators, ashpit doors, etc., and if the available areas are too small, the velocity of the air passing through them must be artificially increased.

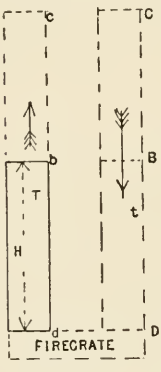


Fig. 87.
Funnel Draught.

Funnel Draught.—A fire, whether of coal or other fuel, must be supplied continuously with oxygen for the continuous combustion or burning of the fuel. This oxygen is derived from atmospheric air, as already stated, and it is now proposed to investigate the natural cause which continues the supply of air. If there are two bodies of fluid in separate chambers but of differing densities, and originally at the same level, the heavier fluid, when a connection is opened between the lower ends of the two chambers, gradually displaces the lighter by pushing it upward. The rate of flow between the two chambers (neglecting frictional resistances) is governed by the head, or difference in density multiplied by the height of the column. When a common fire is burning, the *natural draught*, or rate of flow up the chimney, is caused by the convection current produced by a difference of pressure forcing through the fire a continuous supply of fresh air to the fuel.

In Fig. 87, let DC (= *dc*) represent the height of the atmosphere above the firegrate.

,, DB (= *db*) represent the height of funnel above the firegrate.

And let A = area of each of the columns DC and *dc*
= area of funnel.

Then, weight of air in column DC = weight in columns (BC + DB),
and ,, ,, ,, dc = ,, ,, (bc and db),
in which the weight in column BC = weight in column bc.
Difference in weight = weight in column DB − weight in column db.

If there be no wind or cross current at the top of the chimney or funnel, it is possible that the weight of the column bc may be less than that of BC, because of its higher temperature ; but for all practical purposes this difference is negligible. It is concluded, therefore, that the velocity, or rate of flow, inside a chimney or funnel is due to the head of pressure created by the difference in weight of two columns of ascending and descending air or gases of the same height and area as the ascending column. For all practical purposes, *but only* when both are at the *same temperature* and pressure, the weight of the products of combustion per cubic foot is equal to that of a cubic foot of atmospheric air.

Let the height of funnel above the firegrate be H in feet.
area of funnel opening ,, A in sq. ft.
absolute temperature of funnel gases ,, T_0.
,, ,, air-supply ,, t_0.
velocity of gases inside funnel ,, V in feet per second.
weight of gases passing through funnel ,, W in lb. per second.
,, 1 cubic foot of funnel gases ,, u in lb.
,, ,, ,, air-supply ,, w in lb.
pressure of atmosphere at top of funnel ,, q in lb. per 1 sq. ft.

Then weight of gases inside funnel $= A \,.\, H \,.\, u$.
pressure outside funnel, at firegrate level $= q + A \,.\, H \,.\, w$.
pressure inside funnel $= q + A \,.\, H \,.\, u$.

Difference of pressure outside and inside when gases are not in motion

$$= A \,.\, H \,(w - u) ;$$

but in practice the gases are in motion upward, and the force producing motion = M . V =

$$\text{Mass} \times \text{velocity} = \frac{W}{g} \times V = A \,.\, H \,.\, (w - u) \qquad (1).$$

In which $W = V \,.\, A \,.\, u$.

Therefore $$\frac{V^2 \,.\, A \,.\, u}{g} = A \,.\, H \,.\, (w - u),$$

and $$V^2 = \frac{(w - u)}{u} \,.\, g \,.\, H \qquad . \qquad . \qquad (2).$$

Now, 1 cubic foot of air at 493° abs. F. weighs 0·08073 lb.

Therefore $$w = \frac{493 \times 0{\cdot}08073}{t_0}$$ (from Charles' Law, $P \cdot V = c \cdot T_0$),

and $$u = \frac{493 \times 0{\cdot}08073}{T_0}.$$

From which $$\frac{w-u}{u} = \frac{T_0 - t_0}{t_0} \quad . \quad . \quad . \quad (3).$$

Substituting in (2),

$$V = \sqrt{\frac{T_0 - t_0}{t_0} \cdot g \cdot H} \quad . \quad . \quad . \quad . \quad (4).$$

Example.—When $T_0 = (585 + 461) = 1046°$ abs. F.,

and $t_0 = 62 + 461 = 523°$ abs. F.,

then $T_0 = 2 \cdot t_0$

and $$V = \sqrt{g \cdot H} \quad . \quad . \quad . \quad . \quad (5).$$

In many large ships H = 82 feet, and, therefore, with the above temperatures, V = 51 feet per second ; but it should be clearly understood that this is only a relative value, and does not take into account the frictional and other resistances which may reduce the velocity by more than one-half.

From certain calculations, into which it is not necessary to enter, it can be shown that the *weight of gases* which can be discharged per second through the funnel is a maximum, when $\frac{T_0}{t_0} = \frac{25}{12}$, approximately.

Thus, if $t = 100°$ F., which is a common temperature, the maximum weight of gases is discharged when $T = 707°$ F. $(= 1168°$ abs. F.). Notwithstanding this fact, it is possible in practice to increase the rate of combustion and quantity of coal burnt by increasing the funnel temperature, because with the increased velocity so produced a less weight of air may be required per 1 lb. of coal. This method, however, increases the funnel loss, and in addition decreases the boiler efficiency from incomplete combustion, increased radiation, and inferior quality of steam.

Damaged Funnel.—If the funnel is damaged and penetrated, its virtual height, H, is decreased, and the draught, otherwise created by it, is partially or wholly destroyed above the height of the lowest hole. Small holes have, of course, a less effect than large ones, and the proportion of the draught destroyed depends on their position and on their area relative to that of the funnel passage-way. In the absence

of sufficient forcing arrangements, and after the funnels are damaged, the rate of evaporation, on which the engine power depends, can only be maintained by raising the funnel temperature. In one instance this is stated to have risen to 1800° F., and the coal consumption to have been doubled for the same power.

Mechanical Draught.—For naval purposes an increased rate of combustion (produced by an increased total supply of air) is obtained by fitting some mechanical means of assisting the draught, which is commonly known as forced draught, and which has already been described. It is really a system for assisting the draught created by the funnel, and can be rendered in two ways: either by (1) increasing the *plenum*, or excess of pressure above that of the atmosphere, by compressing the air before it reaches the firegrate, as in the closed-stokehold or closed-ashpit systems; or by (2) increasing the *vacuum*, or reducing the pressure inside the funnel still further below that of the atmosphere by increasing the velocity of the ascending column. The differences of pressure, whether of plenum or vacuum, are measured by a water barometer (commonly known as the air-pressure gauge), in which the atmospheric pressure of 14·72 lb. per square inch supports a head of water of 34 feet.

$$\left.\begin{array}{l}\text{Thus, 1 inch (by gauge)}\\ \text{head of water}\end{array}\right\} = \frac{14{\cdot}72 \times 144}{34 \times 12} \text{ lb. pressure per 1 sq. ft.}$$

$$= 5{\cdot}19 \text{ lb., and the force producing motion}$$

$$= 5{\cdot}19 \times \text{A},$$

$$\text{which from (1)} \qquad = \text{A} \,.\, \text{H} \,.\, (w - u) \text{ evidently.}$$

$$\text{And, therefore, H} \,.\, (w - u) = 5{\cdot}19$$

$$\text{H} = \frac{5{\cdot}19}{w - u} \quad . \quad . \quad . \quad . \quad (6);$$

$$\text{in which } (w - u) = \frac{493\ (\text{T}_0 - t_0) \times 0{\cdot}08073}{\text{T}_0 \cdot t_0} \text{ from (3).}$$

For example, taking the same temperatures as in (5), and substituting the values in (6), then H = 140 feet, nearly.

That is, under normal conditions the draught created by 1 inch head of water (by gauge) is equivalent to that created by a funnel of 140 feet in height above the grate; and also, under any similar conditions of temperature it should be noticed that the head of water *varies directly* as the equivalent height.

Equivalent Height of Funnel.—From practical experience with a well-stoked fire and with the usual air-supply and funnel-passage

areas, it is known that for half an inch of water-pressure, as measured on the gauge, about $12\frac{1}{2}$ lb. of coal can be burnt per 1 sq. foot of grate ; and, therefore, in actual practice with efficient stoking, the pressures should very nearly agree with the rates of combustion given in the following table :

Rate of Combustion in lb. of Coal per 1 sq. foot of Firegrate.	Total Difference of Pressure in inches of Water.	Equivalent Height of Funnel above Firegrate, in feet.
5	0·2	28
10	0·4	56
15	0·6	84
20	0·8	112
25	1·0	140
30	1·2	168
35	1·4	196
40	1·6	224
45	1·8	250
50	2·0	280

The total difference quoted above is that between the pressure below the fire and that in the base of the funnel ; or if two gauges are used, it is the sum of the differences between the atmosphere and the ashpit and between the atmosphere and the base of the funnel. The latter difference is sometimes very small although the height of funnel is great, while the former is very great because so much depends on the stoker in these matters. A well-stoked fire of moderate thickness, say 6 to 8 inches, absorbs from one-half to seven-tenths of the total difference in pressure, which increases with the thickness of the fire. With a fire 24 inches thick, about 5 inches of water-pressure would be required to burn 50 lb. per sq. ft. of grate, the funnel temperature would be about 1400° F., and flaming would be extremely probable ; whereas, with a proper thickness, 2 inches of pressure would suffice, the funnel temperature would probably not exceed 700° F., and there would be no probability of flaming.

In ships with a funnel height of 82 feet and upwards, there should be no difficulty in burning 15 to 20 lb. of coal per sq. ft. of grate without mechanical assistance ; but in destroyers and other small vessels where the funnel is short—perhaps not more than 20 feet—and high rates of combustion are necessary to produce great power for weight and space, forced draught is imperative. In the latter cases the height of the funnel may be neglected when designing the forcing arrangements, and means should be provided to produce pressures

somewhat greater than those shown in the above table for the proposed rate of combustion.

The loss of draught created by funnel damage has already been pointed out, and it thus may be stated that *for all warships* forced draught is necessary, and that there should be a moderately large reserve of power in these forcing arrangements.

Relative Value of Heating Surface.—For water-tube boilers, in which the passage area is nearly constant, or only slightly contracted by baffling, the tubes next the fire absorb about one-third of the total heat given out to the boiler from the fires ; the remaining tubes absorb nearly one-half in decreasing proportion, and the funnel and other losses amount to about one-fifth at ordinary rates of combustion. With a greatly accelerated draught the funnel temperature and loss are somewhat increased, and the quantity of heat absorbed by the tubes more remote from the fire might be proportionately increased, because of the greater difference of temperature produced between the gases and the water contained within these tubes ; but the quickness of movement of the gases probably causes an actual reduction. For rough practical purposes the following proportions, with a boiler of the Yarrow type, may probably be fairly approximate (see also Fig. 47) :

Rate of Combustion per sq. ft. of Grate.	Percentage of Heat absorbed by each double row of Tubes.							Funnel and other Losses.	Efficiency.
lb.	1st.	2nd.	3rd.	4th.	5th.	6th.	7th.	Per cent.	Per cent.
24 (coal)	30	12	9	8	7	6	5	23	77
36 (coal)	27	11	8	7	6	5	4	32	68
48 (coal)	25	10	7	6	5	4	3	40	60

Furnace Temperatures.—These can only be obtained with any accuracy from careful experiment with proper instruments, but it is possible to obtain a fair approximation by calculation from observations made in previous experiments.

In a furnace the combustion of good Welsh coal produces about 14,400 b.t. units per 1 lb. of coal burnt, and this heat is distributed among about 20 lb. of gases, made up of the products of combustion and the excess of air supplied, with a specific heat of, say, 0·24.

The rise in temperature is in this case about—

$$\frac{14400}{20 \times 0{\cdot}24} = 3000^\circ \text{ Fahr.}$$

If only the *minimum* amount, about 11·5 lb., of air is supplied, and complete combustion is obtained, the **maximum** rise is—

$$\frac{14400}{12{\cdot}5 \times 0{\cdot}24} = 4800^\circ \text{ Fahr.}$$

Assuming a fire of about 6 inches in thickness, and that the air is supplied at a temperature of 100° F., the approximate temperatures may be as below :

Position.	Degrees Fahr.	Remarks.
Below the grate .	100	Air only, not heated before supply.
At 1 inch above .	1600	Excess of air, about 25 lb. per 1 lb. of fuel.
„ 2 inches above	3000	„ „ „ 6 lb. per 1 lb. of fuel.
„ 3 „ „	4000	Complete combustion practically.
„ 4 „ „	3000	Lack of air, reducing CO_2 to CO, partially.
„ 5 „ „	2500	„ „ „ CO_2 to CO, completely.
„ 6 „ „	3100	More air supplied above the fire, burning CO to CO_2.

Heat lost up the Funnel.—In calculating this quantity it is necessary to know—

1. The heat given out by combustion : say 14,400 b.t. units per 1 lb. of fuel.
2. The temperature of air supplied : say 100°.
3. The weight of air supplied per 1 lb. of fuel : say 19 lb.
4. The temperature of the gases leaving the funnel : say 600°.
5. The specific heat of the escaping gases, *i.e.* the amount of heat contained in 1 lb. by weight : say 0·24 (water = 1).

For example, take the figures stated above ; then—

$$\text{Funnel loss} = 20 \times 0{\cdot}24 \times (600 - 100)$$
$$= 2400 \text{ b.t. units per 1 lb. of fuel.}$$

Or, expressed as a fraction of the heat supplied—

$$\frac{2400}{14400} = \frac{1}{6} = 16{\cdot}66 \text{ per cent.}$$

If there be no other boiler loss, such as radiation, then the efficiency of the boiler is—

$$100 - 16{\cdot}66 = 83{\cdot}33 \text{ per cent,}$$

and this must be the *maximum* possible under the conditions stated.

If the escaping gases are used to heat the air supplied for combus-

tion it is possible to reduce the funnel loss. Mechanical draught must, however, be generally employed, or the draught will not be sufficient to keep the air and gases in motion. The power used to obtain this must also be counted as a loss, but such power when properly applied more than repays the expenditure. The theoretical loss can be reduced to about 5 per cent under exceptional conditions, that is, when the temperature of the air supplied is approximately equal to that of the escaping gases.

The economiser reduces the funnel loss by taking heat from the gases and imparting it to the feed water contained in it.

Howden's system of closed ashpits and hot-air supply approaches very closely to the highest practical attainment of the least possible funnel loss.

Velocity of Funnel Gases.—These velocities vary considerably in practice for different types of boilers, due to the varying resistances offered by the change in direction and varying passage areas; only an average result can be obtained by calculation, and this is merely an approximation of the actual velocities.

(*a*) The passage-ways allowed in practice for cylindrical boilers are calculated from the grate surface, and are generally a fraction of it; thus—

One-fifth	the grate area	is allowed	over the fire bridge.
One-sixth	,,	,,	through the tubes, not including ferrules.
One-seventh	,,	,,	,, uptake.
One-sixth	,,	,,	,, armour gratings.
One-eighth	,,	,,	,, funnel.

(For water-tube boilers, about 5 per cent greater areas are allowed.)

These areas must either be known or assumed; and also—

(*b*) The weight of air supplied per 1 lb. of fuel, say 19 lb.

(*c*) The heat given out respectively to the furnace, combustion chamber, and tubes.

(*d*) The funnel temperature and loss, previously estimated or observed.

If 14,400 b.t. units are given out, experiments tend to show that—

The furnace will absorb about	6000	b.t. units.
combustion chamber will absorb about	3000	,,
tubes will absorb about	3000	,,
funnel loss will be	2400	,,

The heat passing over the bridge = 14,400 − 6000 = 8400 b.t. units,

distributed among 20 lb. of gases, of specific heat of about 0·24; therefore—

$$\text{Rise in temperature at bridge} = \frac{8400}{20 \times 0{\cdot}24} = 1750^\circ;$$

and if the air is supplied at 100°, then—

$$\text{Temperature at bridge} = 1750 + 100 = 1850^\circ \text{ Fahr.}$$

Similarly, the temperature in the combustion chamber lowered by the heat given out—

$$= \frac{3000}{20 \times 0{\cdot}24} = 625^\circ;$$

therefore—

$$\text{Temperature at entrance to tubes} = 1850 - 625 = 1225^\circ.$$

And again, similarly—

$$\text{Temperature at exit from tubes} = 1225 - 625 = 600^\circ.$$

After the temperatures have been found at the various parts, it is possible to estimate fairly accurately the volumes of the gases passing through the passage-ways. But the rate of combustion must also be known. Assume it to be 30 lb. per square foot of grate per hour; then—

Coal burnt per minute $= \frac{30}{60} = 0{\cdot}5$ lb. per square foot of grate.

Weight of gas produced $= 20 \times 0{\cdot}5 = 10$ lb.

And the volume of this gas at 32° Fahr. $= 10 \times 12{\cdot}38 = 123{\cdot}8$ cubic feet per minute per square foot of grate

(assuming that 12·38 cubic feet go to 1 lb. of gas at 32° F.).

From the law of Charles and Gay-Lussac, $pv = ct$.

$$\text{At the bridge this volume becomes} = 123{\cdot}8 \times \frac{1850 + 461}{493}$$

$$= 578 \text{ cubic feet.}$$

$$\text{At the entrance to the tubes} \quad ,, \quad = 123{\cdot}8 \times \frac{1225 \times 461}{493}$$

$$= 421 \text{ cubic feet.}$$

$$\text{At the exit from the tubes} \quad ,, \quad = 123{\cdot}8 \times \frac{625 \times 461}{493}$$

$$= 272 \text{ cubic feet.}$$

At the bridge the passage area is one-fifth of the grate area, and—

The **velocity** over firebridge	$=578\times5=2890$	ft. per min.	$=48$	ft. per sec.
,, entering tubes	$=421\times6=2426$	,,	$=40$	,,
,, exit from tubes	$=272\times7=1904$	,,	$=32$	,,
,, up funnel	$=272\times8=2176$	,,	$=36$	,,

If the funnel height and temperature are known, the theoretical velocity of the gases due to these qualities can be estimated; but the resistances reduce this very considerably. The actual velocity in practice is only about one-half the estimated value when natural or funnel draught is used.

Analysis of Flue Gases.—Orsat's apparatus (Fig. 88), of which

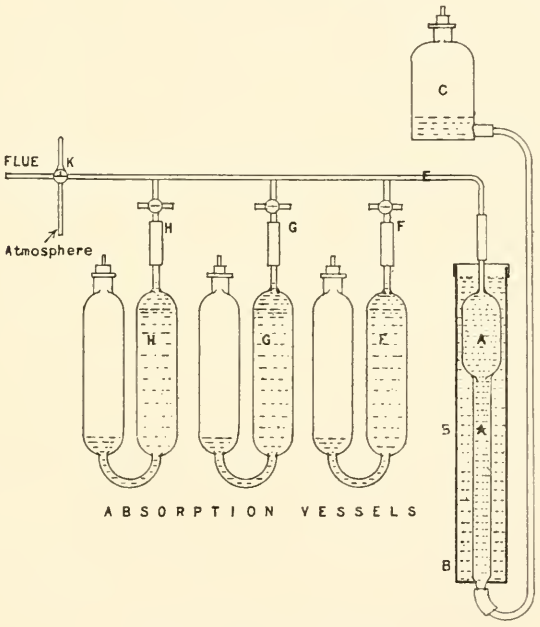


FIG. 88.—Analysis of Flue Gases.

there are many modifications, consists essentially of (*a*) a graduated measuring tube *A*, contained within an outer bottle *B*, which forms a water jacket for maintaining a nearly constant temperature; (*b*) a level bottle *C*, filled about two-thirds full with water in the first instance—the top is partially closed, but the bottom is connected through a flexible capillary tube with the bottom of *A*; (*c*) a glass capillary tube *E*, connecting the top of *A* with three U-shaped absorption vessels, *F*, *G*, and *H*, through stop-cocks, and with the flue gases

or with the atmosphere (and a pump) through the three-way cock K; (d) the vessels F, G, and H are each packed with bundles of glass tubes to increase the absorption surface, and then filled rather more than half-full respectively with a solution of caustic potash of specific gravity 1·20 (for F), an alkaline solution of pyrogallol, consisting of the same solution as in F, but with from 15 to 25 grammes added of pyrogallol (for G), and a concentrated solution of cuprous chloride in hydrochloric acid (for H). Copper spirals are introduced into the small glass tubes in H to maintain the solution in an unchanged state. The solution in F is used to absorb carbon dioxide, in G to absorb oxygen, and in H to absorb carbon monoxide; and it is important that the gas for analysis should be introduced into the absorption vessels in the order named.

By raising or lowering, the level bottle C can be used as a pump, and the method of testing is as follows: Raise C and open K to the atmosphere, and thus allow A to fill with water up to the capillary part. Shut off K from A, and open the cock F, then by lowering C the water runs back into it, and draws up the level of the solution in F until it reaches the mark on the capillary neck, when F should be shut off. The levels in G and H are raised in the same manner. The cocks F, G, and H being then closed, C is again raised and A refilled; then K is shut off from A and connected with the flue gas-pipe and the atmosphere.

By using a jet or india-rubber pump the flue gas-pipe is exhausted of air and at the same time filled with flue gas. Next, the cock K is turned so that the flue gas is in open connection with A. This stage of the operations is indicated in the figure. Lower C and allow the flue gas to flow into A until the water-level reaches zero mark near the bottom of A. The measuring tube A should now contain exactly 100 cubic centimetres of flue gas, the quantity contained in the capillary E being relatively so small that it need not be taken into account. To ensure constant (atmosphere) pressure during measurement, the tube E should be connected with the atmosphere after refilling A, and after each absorption next described.

Open F, and again raise C, when the gas flows into F, and the carbon dioxide contained in it is absorbed by the solution. Lower C until the solution in F reaches the former zero (upper) level, and read off the quantity of gas in A. The difference between this amount and the original 100 c.c. is the percentage of CO_2. The process is now repeated to obtain the percentage of O and CO in turn; but the latter

is generally very small, may usually be neglected, and is difficult to obtain with accuracy. The remainder volume represents very nearly the percentage of nitrogen. No account is taken of the hydrogen contained in the water vapour of the gases, which with gaseous coals and liquid may be considerable.

For testing boiler efficiencies and for power stations on land, an elaboration of this apparatus is frequently used, and by its means the percentage of CO_2 is automatically recorded. This arrangement is sometimes adopted for ordinary marine work. For good Welsh coal the percentage of CO_2 should be *chemically* about 20 per cent; but actually, owing to the necessary excess supply of air, is about 12 per cent. In good practice the maximum is about 15 per cent; above this it points to too little air-supply, the presence of unburnt CO, and possible flaming. The minimum should not be less than 10 per cent; below this the supply of air is too great, and it points to bad stoking with holes in the fire.

The "W. R." Combustion Indicator.—The following table, based on exhaustive experiments and calculations, shows the percentage of coal wasted through guesswork firing or stoking:

Fuel wasted, per cent.	50,	35,	27,	23,	21,	20.
CO_2, per cent.	5,	7,	9,	11,	13,	14.

Computed on a funnel temperature of 600° Fahr.

By increasing the proportion of CO_2 in the funnel gases from 7 to 14 per cent, a saving is effected of 15 per cent in the fuel consumed. It is necessary to have some form of CO_2 recorder on a boiler plant to get the most economical results, and this indicator effects economy in the simplest manner. It is claimed for the apparatus, which is based on a new principle discovered during the war, that it is cheap, simple, automatic, efficient, fireproof, foolproof, and durable, and that it gives an accuracy within one per cent of the CO_2 present in the flue gases.

In Fig. 89 a sketch of the apparatus shows the action diagrammatically. A minute steam jet continuously aspirates gases from the smoke box at about the level of the top of the boiler, through a one-inch iron or copper pipe and a filter *F* in close proximity. The gases then pass through a lead pipe, half inch diameter, and not less than 30 feet long, to the instrument, which consists of a chamber containing a porous pot in which a cartridge of an absorbent reagent (solid caustic potash) is placed. The chamber is fitted with a water barometer, the top end of which is connected with the interior of the porous pot, and

the bottom end with the lower part of the chamber between it and the pot. Some of the gases penetrate to the interior of the porous pot and are absorbed by the reagent, with the result that a partial vacuum is formed and water is forced up the pipe *G*, which is provided with a

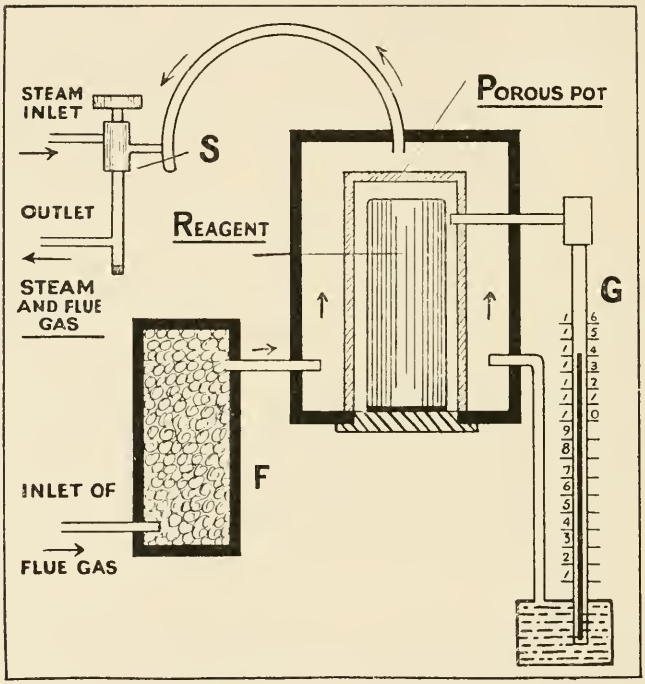


Fig. 89.—"W. R." Combustion Indicator.

scale showing the percentage of CO_2. The reagent requires renewal every alternate day, and the cartridge requires about one minute to renew. The steam injector consumes less than one pint of water per hour, and with forced or induced draught it can be dispensed with. The steam should be as dry as possible.

CHAPTER XII

COMBUSTION OF LIQUID FUEL

Liquid Fuels (Spirits and Oils).—In comparing the relative value of spirituous and oil fuels it is necessary to know certain *characteristics* of their nature, such as :

1. *Specific gravity or density*, which is the ratio of the weight of 1 gallon of fuel to the weight of 1 gallon of fresh water at the same temperature and under the same pressure ; usually 60° F. or 15° C. and atmospheric pressure (29·92 inches of mercury) are the standards at which the weight of 1 gallon of fresh water is 10 lb. Thus :

1 gallon of petrol (density 0·70) weighs 7 lb.
1 gallon of liquid fuel (density 0·90) weighs 9 lb.

2. *Vaporising-point*, which is the temperature at which the liquid gives off vapour in sufficiently large quantity to form an inflammable (*i.e.* carburetted) mixture with air. Alcohol, benzol, and the lighter petroleum spirits, such as petrol, will give off such vapour in varying degrees at any temperature above 32° F. (freezing-point). The vaporising-point is always below the flash-point.

3. *Flash-point* is the temperature at which the spirit or oil, when heated slowly in a cup, begins to give off a sufficient quantity of inflammable vapour to burn with a momentary blue flash when a small test flame is brought into contact with the *vapour*. The general standard is the *Abel closed test*, in which the oil is heated in a *closed* cup, and either the " Pensky-Marten " or the " Gray " tester is generally used for this purpose.

4. *Burning-point* is the temperature at which the *liquid* takes fire and *continues* to burn. (An ordinary example of this is the heating in a spoon and burning of brandy to be poured over the Christmas pudding.)

5. *Boiling-point* (which must not be confused with the vaporising-

point) is the temperature at which ebullition or boiling begins under atmospheric pressure, thus changing the state of a liquid to that of a gas. (Petrol, at ordinary atmospheric temperature, *vaporises* at the rate of about $2\frac{1}{4}$ oz. per square foot of exposed surface, and methylated spirits vaporise at almost one-third this rate ; this is vaporisation, not boiling.)

The following table shows approximately the various characteristics of the ordinary spirituous and oil fuels :

	Density at 60° F.	Vaporising-point. Deg. F.	Flashing-point. Deg. F.	Burning-point. Deg. F.	Boiling-point. Deg. F.
Alcohol (before denaturising)	0·76	..	140	..	173
Wood Alcohol	0·77	..	..	..	151
Benzol	0·883	..	..	..	198
Petroleum products :					
Petrol	0·70	0° about	0 to 10	..	163
Naphtha	0·75	..	120	..	..
Paraffin	0·82	..	73 to 80	..	300
Kerosine	0·78	..	77 to 122	..	290
Russoline	0·824	..	82	..	303
Shale Oil :					
Broxburn	0·81	..	125	..	329
Liquid Fuels :					
German Navy	0·89	..	150	..	445
U.S. Navy	0·90	135	180	210	500
British Navy	0·9	140	175	222	500
,, ,, Shale Oil	0·86	..	200	..	..

The great differences exhibited by these characteristics for fuels of various natural origin are likely to mislead, and consequently the author has preferred not to quote all of them, but those remaining are sufficient to show the general tendency and relative differences which appear to follow no known law of exact relationship. The most important characteristic is the flash-point, which is of the highest importance to safe storage and usage in confined spaces. The flash-point should be specified for all fuel oils, especially those of the kerosine series, which are commercially known as naphtha, benzoline, benzine, russoline, and the numerous brands of illuminating oils.

The boiling-point of petroleum products, although fairly definite with regard to the temperature at which boiling begins, is not very clearly defined with regard to the *range* of temperature over which distillation continues, which in some instances is very wide. The wider the range the greater are the difficulties in obtaining good combustion, both under boilers and in internal combustion engines, from

sooting up of the valves and cylinder clearance and exhaust spaces. In the selection of a fuel suitable for internal combustion engines a low range of temperature of distillation is of the greatest importance to continuous successful operation.

Professor Spooner states this range for benzol at 91°, from 198 to 289, and for petrol at 187°, from 163 to 350, presumably under normal atmospheric pressure. The range is due to the various fractional constituents of the liquid.

6. *Viscosity* is a measure of the fluidity of the liquid, and an apparatus invented by Sir Boverton Redwood is used for the purpose. It consists of a small cup in which the liquid can be heated and steadily maintained at any required temperature. The time required *in seconds* for a measured quantity (1000 grains) of the liquid to escape through a hole of standard size in the bottom of the cup is termed the viscosity of the liquid. Various standard temperatures are used ranging from 70° F. to 100° F.; the higher the temperature the less becomes the viscosity. The tests for viscosity are much used for lubricating oils and generally for all petroleum products. The standard for viscosity is genuine sperm oil, which has a viscosity of 100 at a temperature of 70°; this determines the size of the hole, which is drilled through a small piece of agate fitted into the bottom of the cup.

Shale Oils. — In Scotland, particularly near Edinburgh, in Australasia, and elsewhere, lighting paraffin and other fuel oils are obtained by distillation from oil shale. Their chemical composition and other characteristics are similar to oils distilled from petroleum. Scotch shale also produces about 12½ per cent of paraffin wax, which is made into candles or used for other purposes.

The following grades of oil are being produced by the British Australian Oil Company. They are characteristic of those usually produced in the refining of shale and mineral petroleum oils. In the order stated below the lightest and most easily evaporated oils are first produced: Benzoline, motor spirit, benzine, naphtha, burning oil (1st quality), burning oil (2nd quality), gas oil, fuel oil, lubricating oil, heavy lubricating oil, cylinder oil, residual oil, wood-preserving oils. Paraffin wax and greases are expressed from fuel and heavier oils; coke and sulphate of ammonia are produced in the refining processes necessary to the production of lubricating oils. About 21 to 22 lb. of sulphate of ammonia are produced from 1 ton of shale.

Petroleum and other Oils.—The use of oil for lighting and

heating purposes is of very ancient origin, and it is only necessary to recall the parable of the Ten Virgins to notice that oil was widely known in the early days of the Christian era, although it is probable that at that time the oil was of vegetable origin and obtained from rapeseed, nuts, and olives; but if petroleum oil was not used, the "City of Eternal Fire" (Baku) and its Fire Temple were well known to travellers of the olden time, and many other sources were known in the Ionian Islands, Burmah, and the Eastern Archipelago. Within the last century, and particularly during the latter half of it, oil has been discovered in almost all parts of the world, and there appears to be a chain of petroleum-bearing earth which follows more or less closely the chain of the volcanic regions. In the geological exploration for petroleum the recurrence of fossil strata is probably the most interesting and convincing proof of the existence of petroleum. Nearly every oil-producing company has its geological museum, and by collecting fossils at certain known points the probable presence and depth of the oil-bearing strata are determined with an almost uncanny certainty.

In a few cases the strata reach the surface in pitch lakes, of which the island of Trinidad is a well-known instance. Tar, pitch, naphtha, petroleum, and bitumen all belong to the same family, and it is highly probable that coal is nearly allied to them from the simple fact that in many countries coal and oil have been found in close proximity. Petroleum and the natural gases evolved from it have frequently been discovered by the presence of an oil film in streams and pools, and in other instances a film of oil has covered wide areas of sea, thus pointing to a subaqueous volcanic origin.

In some cases the well is a spouter, and the piercing of the crust of the reservoir allows the oil to rise many hundreds of feet above the surface of the land, while in others it is necessary to pump the oil to the surface from great depths.

The natural petroleum or *crude oil* obtained from wells, being of a very inflammable character and therefore unsafe, is not usually sold commercially as a fuel. In the early stages of distillation and refining, the lighter gases with low flash-points are given off and condensed again ready for storage; these gaseous oils or spirits, termed "benzene," are generally used as fuel for the refining processes, and thus nothing is wasted.

According to origin and requirements, governed by commercial considerations, the crude product undergoes few or many fractions of distillation, and in this way a large number of spirituous oils—

benzene (already mentioned), petrol, kerosine or paraffin, benzine legally called *heavy* oil (having a flash-point above 73°), and heavy lubricating oils of various higher flash-points and boiling-points—are produced. In each separate fractional product traces of the various lower flash-point products are to be found. To some extent these different traces, which in the early stages may be a large proportion, cause some difficulty in efficient atomising and in burning completely, from the too early vaporisation of the lower flash-point traces. As the temperature of distillation increases, the density of the *residue* also increases, and thus the lighter oils always possess a lower flash-point.

As an example of what may result from fractional distillation, natural oil from Gemsah (Egypt) has a specific gravity of about 0·8175 at 59° F., is of sulphurous odour, inflammable at ordinary temperature, viscous to the touch, and orange-brown in colour with a greenish tinge.

Slow continuous distillation gives the following fractions:

	Specific gravity.	Proportion.
Commercial petrol	0·710	22·6 per cent.
Heavy petrol	0·770	6·5 „
White burning oil	0·809	12·7 „
Ordinary burning oil	0·830	30·4 „
Gas oil	...	14·2 „
Coke	...	4·1 „
Loss	...	9·5 „
		100·0 „

Petroleum residue is generally very thick and sluggish in its movements. Before burning under a boiler it is usually necessary to heat it considerably so as to make it sufficiently fluid, and in addition considerable pressure of air or steam is necessary to atomise it. The rate of combustion is in every case very slow, and this renders it unsuitable for high-power purposes as a boiler fuel, but it is used successfully at low rates of combustion in boilers specially designed for its use. The residue is unsuitable for internal combustion engines, and intermediate refined products of flash-points between 32° and 250° F. are used for this purpose.

Liquid fuel for naval purposes, both in boiler furnaces and in internal combustion engines, is produced by fractional distillation *in vacuo*, and such portions of the distillate are taken as will yield fuels of the required flash-point and cold setting-point. For safe storage (and burning under boilers) the flash-point must be fairly high, and to ensure its easy passage through the small holes of the burners (for boilers) and the fuel inlets (for internal combustion engines), it is

necessary to keep the oil in a perfectly fluid state down to a low temperature. American oils and Burmah oils, which contain 2 and 7½ per cent respectively of wax, are therefore subjected during the refining processes to a low temperature at which the wax can be removed by mechanical separation. Russian oils contain no wax, and the separation process is unnecessary, but as fuels for internal combustion engines they are usually less satisfactory than Burmah and American oils.

Admiralty Specification for Oil Fuel (1912).—A new specification for oil fuel was introduced provisionally in 1912, and supersedes that of 1910. Both specifications refer to *Quality* only :

1. The oil fuel supplied shall consist of liquid hydrocarbons, and may be either—

(*a*) Shale oil ; or
(*b*) Petroleum as may be required ; or
(*c*) A distillate or a residual product of petroleum, and shall comply with the Admiralty requirements as regards flash-point, fluidity at low temperatures, percentage of sulphur, presence of water, acidity, and freedom from impurities.

2. The flash-point shall not be lower than 175° F., close test (Abel or Pensky-Martens). In the case of oils of exceptionally low viscosity, such as distillates from shale, the flash-point must be not less than 200° F.

3. The proportion of sulphur contained in the oil shall not exceed 3 per cent.

4. The oil fuel supplied shall be as free as possible from acid, and in any case the quantity of acid must not exceed 0·05 per cent calculated as oleic acid when tested by shaking up the oil with distilled water, and determining by titration with decinormal alkali the amount of acid extracted by the water, methyl orange being used as indicator.

5. The quantity of water delivered with the oil shall not exceed 0·5 per cent.

6. The viscosity of the oil supplied shall not exceed 2000 seconds for an outflow of 50 cubic centimetres at a temperature of 32° F., as determined by the Redwood Standard Viscometer (Admiralty type for testing oil fuel). (Pending settlement of this specification a viscosity of 1000 seconds was provisionally adopted in 1912.)

7. The oil fuel supplied shall be free from earthy, carbonaceous, or fibrous matter, or other impurities which are likely to choke the burners.

8. The oil fuel shall, if required by the inspecting officer, be strained by being pumped on discharge from the tanks, or tank steamer, through filters of wire gauze having 16 meshes to the inch.

9. The quality and kind of oil shall be fully described. The original source from which the oil has been obtained shall be stated in detail, as

well as the treatment to which it has been subjected, and the place at which it has been treated.

10. The ratio which the oil supplied bears to the original crude oil should also be stated as a percentage.

A note is added to the effect that the modifications made in regard to flash-point, sulphur, acidity, and viscosity in 1912 were made as the result of expert advice and experiment, and further experiments are still proceeding. It is stated that some further modifications in respect of the percentages of sulphur and water, and of the standard of viscosity, have been, and will be admitted within necessary limits, on account of natural variations of oils from different sources of supply.—(*The Times*, Engineering Supplement, August 20, 1913.)

Physical properties and chemical composition of fuels :

Description.	Carbon, per cent.	Hydrogen, per cent.	Oxygen, per cent.	Sulphur and Ash, per cent.	Lower scale calorific value per 1 lb. in B.Th.U.	Specific gravity.
South Wales coal	84·0	5·0	4·0	7·0	14,580	...
Newcastle coal	82·0	5·0	6·0	7·0	14,158	...
Scotch coal	78·5	5·6	9·7	6·2	13,730	...
French coal-tar oil	82·0	7·6	10·4	...	15,250	1·044
German coal-tar oil	89·0	6·7	4·3	...	16,200	1·02
Benzol	90·1	8·0	0·34	1·56	17,300	0·88
Paraffin from brown coal	85·5	11·4	3·1	...	18,270	0·90
Methyl alcohol	41·5	13·0	45·5	...	9,920	0·69
Petroleum spirit (petrol)	84·3	15·7	...	...	20,600	0·70
Russoline	85·8	14·0	0·05	...	19,900	0·82
Crude petroleum	86·0	11·0	1·0	2·0	18,260	0·8–0·925
Burmah natural petroleum	84·0	12·5	3·5	...	18,610	0·875
Russian astaki	85·0	14·0	1·0	...	19,720	0·9

NATURAL FUEL OILS
WITH BROXBURN SHALE DISTILLATE FOR COMPARISON

Place of Origin.	Carbon.	Hydrogen.	Sulphur and Ash.	Oxygen.	Specific gravity.	B.Th.U. per 1 lb.	Calories per kilogram.
Mexico	83·6	11·2	4·0	1·2	0·95	18,000	10,000
Borneo	86·5	11·2	0·1	2·2	0·94	18,415	10,233
Beaumont, Texas	84·63	12·0	1·37	2·0	0·93	18,541	10,302
Galicia	84·3	12·6	3·0	0·1	0·86	18,937	10,525
Russia	86·1	12·3	...	1·6	0·91	18,937	10,525
Java	87·1	12·2	0·3	0·4	0·91	19,024	10,573
Texas	87·22	12·32	0·37	0·09	0·91	19,210	10,676
Broxburn, Distillate	86·4	12·7	0·4	0·5	0·86	19,285	10,718
Parma, Italy	84·0	13·4	1·8	...	0·79	19,328	10,741
Pennsylvania	84·9	13·7	...	1·4	0·89	19,514	10,828
West Virginia	84·3	14·1	...	1·6	0·84	19,640	10,897

In the above table, the *lower* scale calorific value is given both for English and metrical units of heat.

1 b.t.u. = 0·252 calorie. 1 b.t.u. per 1 lb. = 0·554 calories per 1 kilogramme.

1 calorie = 3·968 b.t.u. 1 calorie per 1 kilogramme = 1·8 b.t.u. per 1 lb.

It should be clearly understood that the above figures are results obtained from certain samples, and that other samples might and probably would give different properties and composition. The difference is, however, unlikely to be greater than 1 per cent, and for practical purposes may generally be ignored. Refined petroleum oils intended for lubrication purposes should not contain more than $\frac{3}{4}$ per cent of sulphur and deleterious substances of an acid nature.

Settling or Separation Tanks.—The presence of water, which is very common in small or large quantities in the oil fuel, increases the difficulty of obtaining efficient combustion, and considerably reduces the evaporative power. Every 1 per cent of water contained in the fuel reduces the evaporative power by about 2 per cent. The presence of water is generally indicated by the colour of the flame.[1] With satisfactory combustion the flame is an opaque dazzling white for about 6 inches from the end of the jet, then gradually becomes semi-transparent, then at about mid-length turns violet, and towards the end changes to red. When mixed with a moderate amount of water no violet is observed, and the red becomes darkened and fringed with smoke. If only a small proportion of water is present, instead of the dark red, the end of the flame is white, and this is probably due to acetylene.

A large proportion of the sulphur is removed before its delivery for fuel, and the water is dealt with before delivery to the burner.

The densities of the oil fuel (0·9) and water (1) do not differ very much. The lighter oil does not readily rise to the surface, and it is necessary to accelerate the separation by heating or other means. To do this the fuel is pumped into separation tanks, each containing about twelve hours' supply, or a total of about 20 tons for the ordinary cargo steamer. A coil of piping, heated by exhaust steam, is made to float a few inches below the surface of the oil, and the heat sets up a circulation of the oil and water past it. As the oil and water have differing expansive ratios they separate more rapidly when heated. The separation by heating and circulation leaves the lighter oil at the

[1] Mr. C. E. L. Orde.

surface, while the heavier water gravitates to the bottom of the tank. A gauge glass shows the relative levels of oil and water, and the water can be drawn off from the lower part of the tank and the oil left ready for use. The settling or separation tanks are used in rotation.

When separating the oil from water by heating it, only sufficient temperature for the purpose should be allowed in the heating pipe or coil. Vapour is given off by the fuel at a temperature a little below the flash-point, and any great accumulation is not advisable. Exhaust steam, with a temperature not above 140°, is fairly safe for oil with a flash-point of 160°. It is unnecessary and wasteful to heat the whole of the oil in the separation tank, and a pipe led away from the immediate vicinity of the coil is now frequently used to supply the burner. The oil is then supplied to the burner at a high temperature, and is more easily atomised and gasified.

Atomisers, Burners, or Pulverisers.—The usual method of burning oil fuel is to inject it into the furnace through burners of the atomiser or pulveriser type. The object of atomising is to increase the surface area of the fuel by minutely dividing it into spray of the greatest possible number of particles of the smallest size, and thus allow the air required for combustion to become intimately mixed with the fuel. The finer the division is, approaching very closely to the consistency of a fog when using a good atomiser, the greater the surface area becomes, and the more intimate is the mixture with the air supplied for combustion.

The necessary impelling force used for atomising is generally supplied by—

1. Delivering the oil under pressure, as in the mechanical spray type. (Similar in principle to a fire-hose.)
2. Inducing a flow by a jet of steam, which is frequently superheated. (Similar in principle to a boiler feed water-injector.)
3. Inducing a flow by a jet of compressed air, which is frequently heated.
4. By gasifying the oil in a heater (or vaporiser) and allowing it to escape through the burner.

Any one of these methods can be used either for oil fuel combustion only, or for mixed combustion, when the oil fuel is injected on to the incandescent coal. Various methods are adopted for increasing the efficiency of the combustion: such as heating the air supplied for atomising or for combustion, heating the fuel, superheating the steam

used for atomising, and in some cases a combination of some or all of these is employed.

When the firegrate is retained for mixed combustion the burners are generally fitted near the furnace doors, and in some cases a burner is attached to each door. The burners point slightly downwards, so as to direct the jets on the middle of the fire or thereabouts.

When no firegrate is fitted the arrangements vary considerably. The usual way is to fit the burners as low as possible in the furnace, commensurate with the proper supply of air below the burners, and with obtaining as much room as possible for the completion of the combustion before the hot gases reach the boiler heating surfaces. If the hot gases strike the boiler heating surfaces before combustion is complete, and are cooled by it to a temperature below that necessary for combustion, the combustion is checked and the gases escape only partly consumed, and smoke is produced. In all cases a sufficiency of brickwork is necessary to perfect combustion : it retains the temperature of the furnace at a higher level, and, acting as a heat-accumulator, re-ignites the gases and unconsumed oil, and it also tends to keep the flames in a state of incandescence.

Furnaces, Screens, and Brickwork.—The arrangement of the furnace is of the greatest importance to efficient combustion, and the whole of the available space should be utilised.

When the temperatures of the combustible and the air supplied for combustion are low, the flame is long ; and when these temperatures are sufficiently high, the flame can be shortened to the available length of the furnace, and the time required to complete combustion is also shortened.

The impact of the flame on a cold surface, such as a boiler-plate or water-tube with a temperature of about 400°, checks combustion and produces smoke, as the petroleum deposited does not readily re-ignite. It was usual, therefore, to direct the jet on to a brickwork erection, which, retaining some heat, re-ignited the spray falling on it, and burnt the petroleum away. If the furnace is sufficiently long, it is preferable to leave the flame unrestricted (at present a common practice), but sufficient brickwork must exist to ensure the re-ignition of any fuel dropping on it.

The arrangements used in warships for mixed coal and oil combustion are shown in Figs. 91 and 90, for a Babcock boiler and a cylindrical boiler respectively. Taking Fig. 90 as an example, two burners are fitted, one on each side of the furnace door. Air is supplied

from the stokehold through holes in the circumference of a projection from the end of the furnace, and the amount is regulated by sliding shutters. The burner sprays the oil into the central space of an air cone, with slots cut in its conical face. The slots are made so that the air, entering through the sliding shutters and the slots, is given a

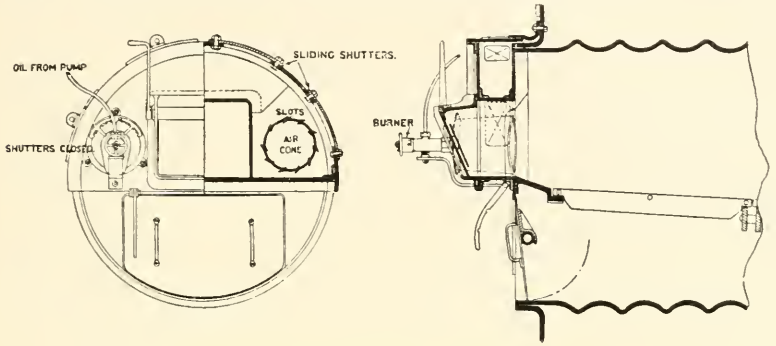


FIG. 90.—Furnace for burning mixed Coal and Oil.

centrifugal spiral motion in an opposite direction to the centrifugal motion of the spray, which is thus very finely divided, and burns completely when the air and oil supply are properly regulated. Air is supplied to the coal fuel through the ashpit in the usual manner.

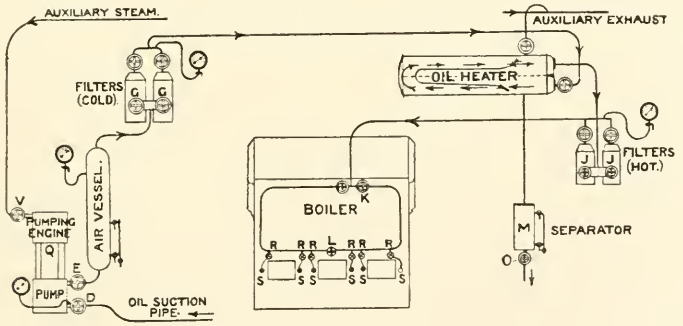


FIG. 91.—Arrangement for burning mixed Coal and Oil

The position and general arrangement of the burners, air cones, sliding shutters, etc., are clearly shown in the figure, and when seen the various details can readily be followed.

The general arrangement for burning oil fuel, either in conjunction with coal, or alone, in a boiler is shown in Fig. 91. The oil enters the oil fuel pump, *Q*, through a suction pipe and valve *D*, and is

delivered under pressure through a valve *E*, and air vessel (of the type shown in Fig. 320), into one or both of the filters (cold) *G*, which are arranged so that either one can be cleaned while the other is in use. From the filter, *G*, the oil enters the oil heater (Fig. 101) which is of the surface type, and warmed preferably by exhaust steam from the closed (auxiliary) exhaust. A temperature of 200° F. is not always obtainable when using exhaust steam for heating, and boiler steam is used, although less economical for this purpose.

The oil when warmed is thinned and flows more easily, and is again filtered through filters *J* (hot), before passing into the burners through valves *K*, *L*, and *R*. *M* is a separator on the heater (steam or water)

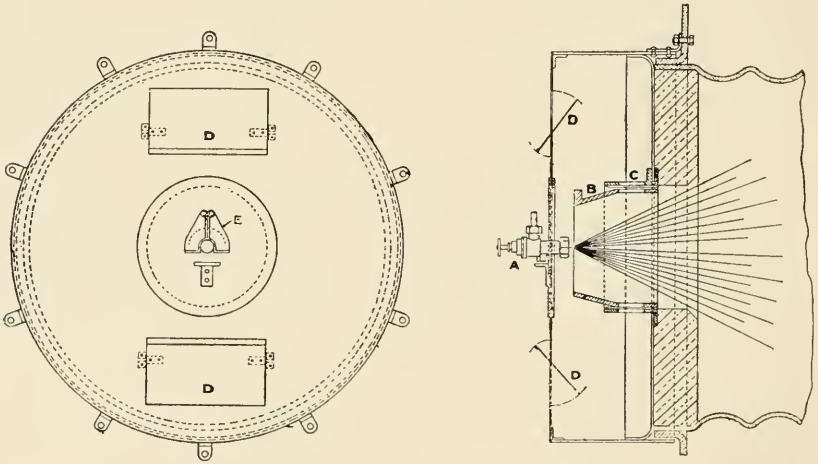


Fig. 92.—Kermode Air Control System for Fuel Oil.

drain, and *O* is the separator drain valve ; when working steadily the highest economy of heating steam is obtained when the drain *O* is set so as to keep the separator about half full of water, as shown by the water-gauge. Pressure-gauges on the oil delivery system show the necessity or otherwise of cleaning the filters, and a difference of 5 lb. and 10 lb. per square inch respectively between the air vessel and cold filter, and between the cold and hot filters, generally obtains when the system is in good working order ; the precise pressure difference is, however, a matter for observation in practice under working conditions with each plant.

In Fig. 92, Kermode's patent air control system is shown as fitted to a cylindrical boiler furnace, but it can be adapted to any other boiler. The automatic louvres *D* ensure a safeguard against back

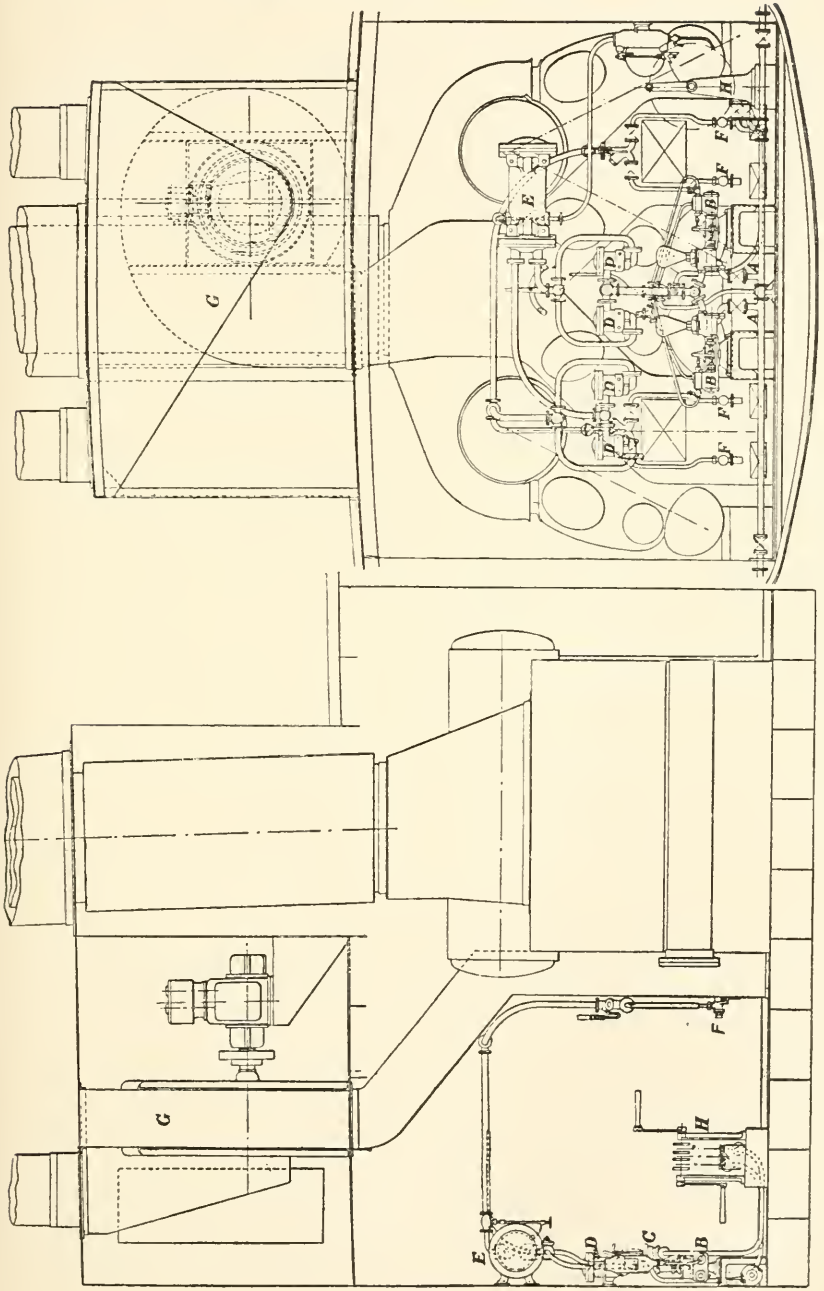


FIG. 93.—Kermode Pressure Jet Oil Fuel System.

firing. *A* is the sprayer or burner which atomises the oil under mechanical pressure. The spray is discharged into the furnace through the Kermode patent air cone, consisting of inner and outer tubes, *B* and *C*, with lanterns. The inner tube *B* can be operated from the stokehold while the burner is at work, and the air opening thus regulated to suit the rate of combustion within certain limits. A sight hole *E* is provided to observe the flame, which should just clear the extreme end of the air cone nearest the furnace by adjusting the louvres *D*, which should be open equally. The burner should point slightly

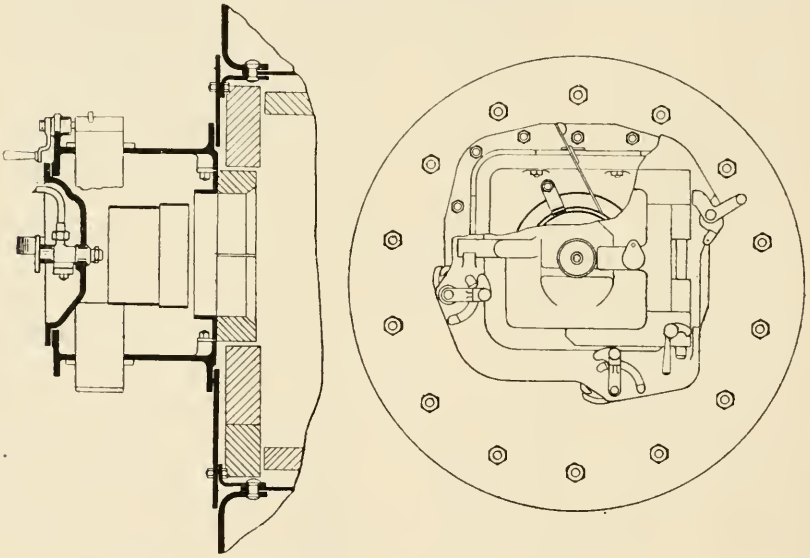


Fig. 94.—Thornycroft Oil Fuel System for Cylindrical Boiler Furnace.

upwards, from 3 to 5 degrees with the horizontal. The inner lantern tube of the air cone can be removed and renewed while the boiler is in operation by removing the front plate of the air control box or casing.

In Fig. 93, Kermode's pressure jet system, as fitted in about sixty French naval vessels and in many others, is shown in connection with two Yarrow boilers as fitted in a vessel of the Dutch navy. The steam-driven oil fuel pumps *BB* take their suction from pipes *AA* and tanks ; an escape valve *C* is fitted to return surplus oil to the suction pipe. The oil is pumped through first and second set of straining filters into the oil fuel heaters *E*, heated by steam, and thence to the burners *FF*.

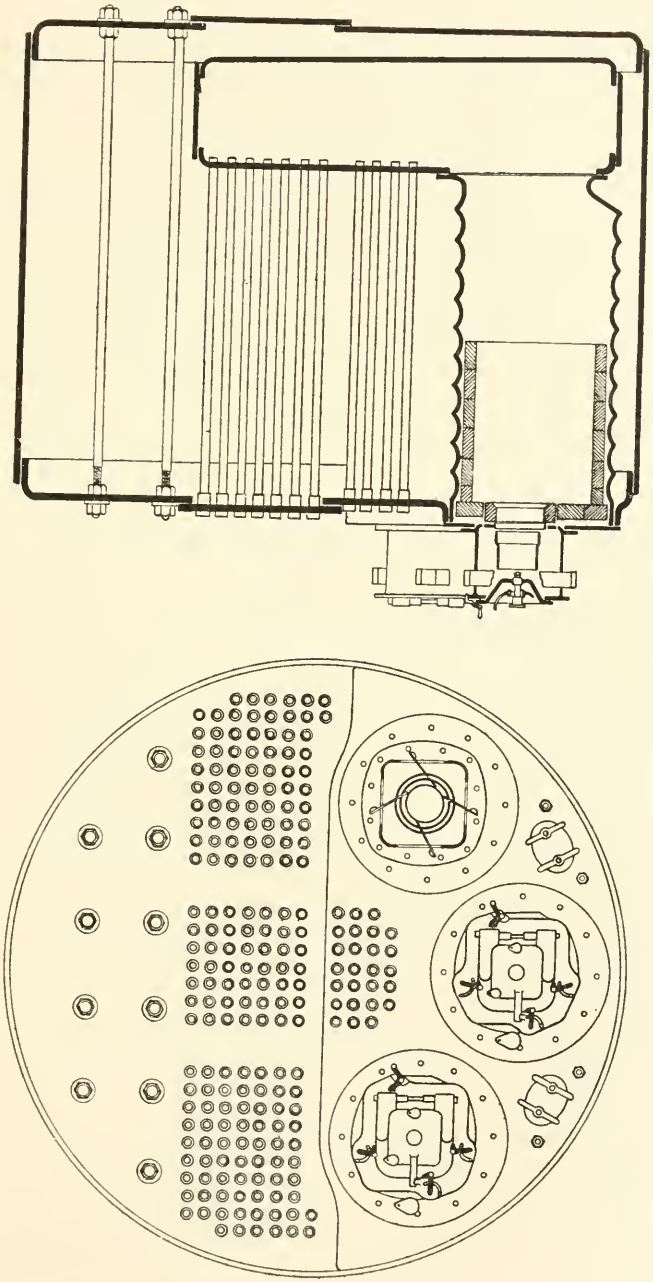


FIG. 95.—Thornycroft Oil Fuel System for Cylindrical Boiler.

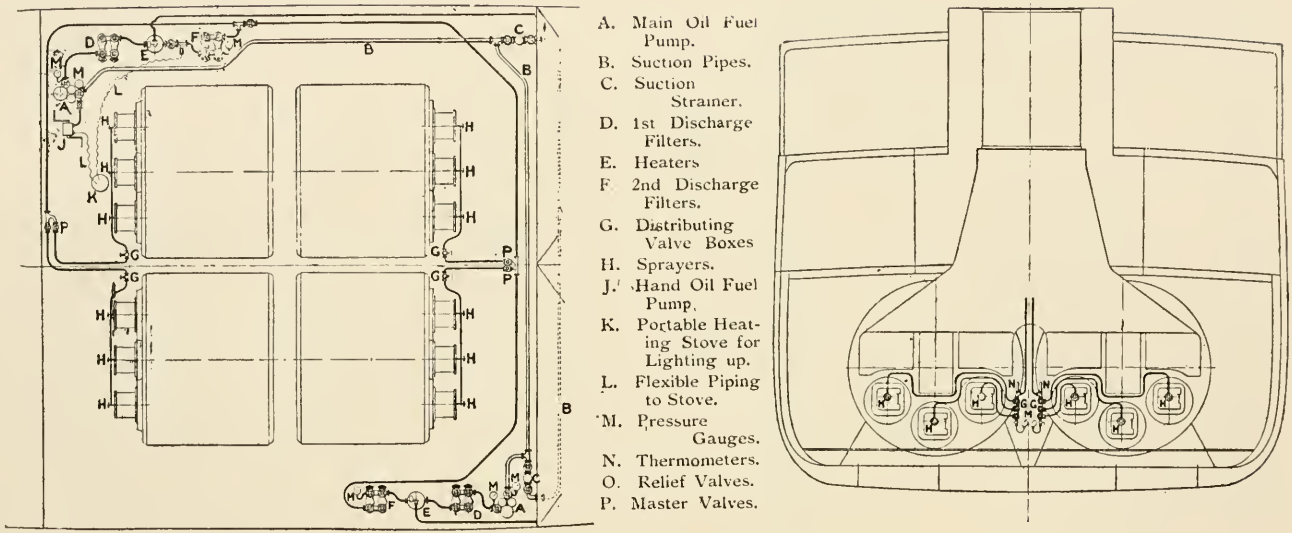


FIG. 96.—Thornycroft Oil Fuel System for Cylindrical Boiler Room.

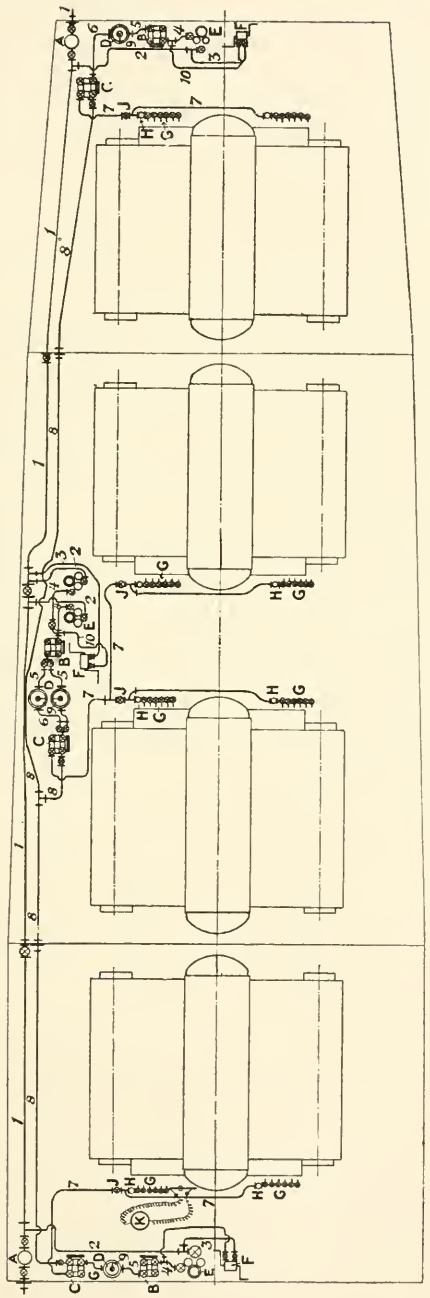


Fig. 97.—Thornycroft Oil Fuel System for Water-tube Boiler Rooms.

Mark.
A. Suction Strainers.
B. First Filters.
C. Second Filters.
D. Heaters.
E. Oil Fuel Pumps.
F. Hand O.F. Pumps.
G. Distributing Boxes.
H. Thermometer Boxes.
J. Control Valves.
K. Portable Heating Stove for Lighting up.

Pipe No.
1. Oil Fuel (O.F.) Suction Range.
2. O.F. Pump Suctions.
3. O.F. Hand Pump Suctions.
4. O.F. Hand Pump Discharge to First Filters.
5. Discharge, First Filters to Heaters.
6. Discharge from Heaters to Second Filters.
7. Discharge from Second Filters to Boilers.
8. Hot Oil Cross Connection.
9. Steam to O.F. Heaters.
10. Discharge from Hand O.F. Pump.

Air is supplied by a forced-draught fan *G*. For initially raising steam a hand pump *H* is fitted.

Figs. 94, 95 and 96 show a Thornycroft oil fuel burning system, as fitted in a mercantile vessel with cylindrical boilers and closed stokehold forced draught. The reference gives a list of the principal fittings, and the pipes can be traced throughout to the burners.

Fig. 97 shows a Thornycroft arrangement of water-tube boilers (40,000 s.h.p.) for burning oil fuel with closed-stokehold forced draught. The reference gives a list of the principal fittings, and from this the general system can be traced.

Rate of Delivery. — The weight or quantity of oil discharged through a mechanical spray burner or nozzle varies as the square root of the pressure or head, because it must be in general agreement with the usual formula—

$$v^2 = 2 \cdot gh.$$

If p = pressure per square inch of oil just previous to discharge, then $v = 8\sqrt{2 \cdot p}$ about, if something be allowed for frictional losses.

On this basis the area A in square inches, and diameter D in inches, of the nozzle bore can be found fairly accurately for a given set of conditions, which is best shown by an example.

EXAMPLE.—The total consumption of fuel oil per hour is 2·59 tons (= 100 cubic feet about), and there are 25 nozzles; find the diameter of the nozzle bore for an oil-pressure of 50 lb. per square inch.

The cubic feet of oil discharged per nozzle per second =

$$\frac{A \times v}{144} = \frac{4}{60 \times 60}$$

$$\text{From which } A = \frac{4 \times 144}{60 \times 60 \times 8\sqrt{2p}}$$

$$= \frac{1}{50\sqrt{2p}}.$$

From which, if $p = 50$, $A = 0{\cdot}002$, and $D = 0{\cdot}05$ inch.

This calculation is only true when the oil has the same consistency as water. In actual practice, although the viscosity of the oil is reduced by heating it to a temperature of nearly 200° F., the pressure must be about *double* that calculated above to produce the same rate of flow as water. Also, efficient atomising of oil is seldom obtainable with a stream greater than $\frac{1}{20}$ inch in diameter, and this or a less diameter is generally adopted for nozzles worked under the pressure system.

A working pressure of between 100 and 150 lb. at the pump is usually adopted, and within these limits some moderate variation in

the rate of flow and burning can be adjusted by regulating the speed of the pump. Further variation is effected by regulating the number of nozzles in use, contingent on each one being worked as near full power and pressure as may be practicable.

Steam Atomiser.—The use of steam for atomising is not economical in marine practice, because it entails a loss of fresh feed water amounting to about 5 per cent of the steam generated in the boiler. The steam tends to reduce the temperature of combustion and to produce smoke, and because all the heat contained in the steam used for atomising eventually passes away with the products of combustion, there is a considerable and unrecoverable loss of heat.

Steam is, however, a convenient means of atomising fuel of high flash-point and considerable density such as the residue from petroleum distillation, and for lower rates of combustion not exceeding the equivalent of 15 to 20 lb. per square foot of grate, it is successfully employed in many locomotives and river craft where a supply of fresh water is easily obtained. Thick oil must be thinned by heating before atomisation, and if steam is used for this purpose it should be superheated or dried by passing it through an efficient separator.

Mechanical Spray or Pressure Jet Burner.—The most common and probably the most efficient system of pulverising and spraying the oil fuel is through a burner or sprayer of the mechanical pressure and jet type, of which there are many makes, but all embody a single principle, *i.e.* some form of whirling or centrifugal action to obtain a fine division of the particles of oil.

Fig. 98 shows the Thornycroft type, of which over $2\frac{1}{4}$ millions of H.P. had been supplied in 1919. The oil fuel enters under pressure at *A*, and fills the space between the outer casing *D* and an inner perforated casing *B*, and into a whirling chamber *K* through two tangential holes. The quantity delivered is governed by the cone-ended spindle *C*, worked by the wheel *F*, and graduated by reference to the indicator *G*. Both edges of the exit holes are tangential to the whirling chamber. The position of the sprayer is clearly shown in Figs. 94 to 97.

Fig. 99 shows the Kermode type of pressure jet burner. The oil fuel enters the burner at *A*, and passes between the outer wall *D* and the inner cylinder *B*, which abuts against the capnut *E*. The end of the cylinder *B* is an exact fit for the outer body *D*, where it abuts against the capnut *E*. A series of grooves are cut in the plug end *B*, which are shown in the sketch marked *Spool* at *H*. These grooves are tangential to the cone end of the spindle *C*, which controls the opening

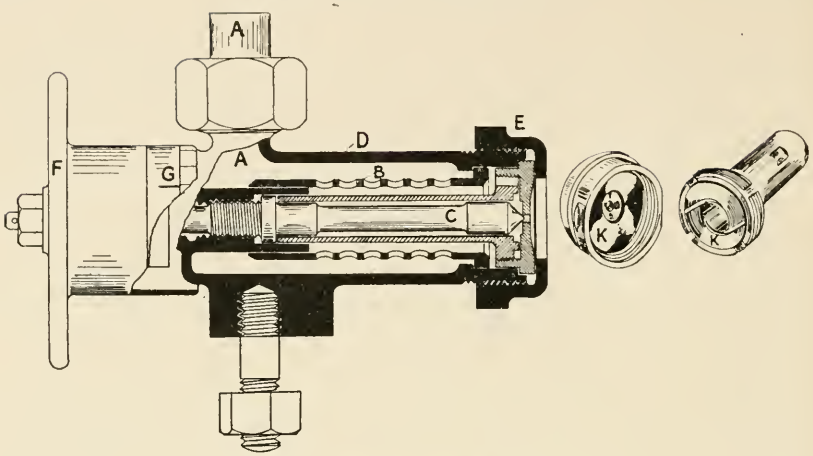


Fig. 98.—Thornycroft Oil Fuel Pressure Jet Sprayer.

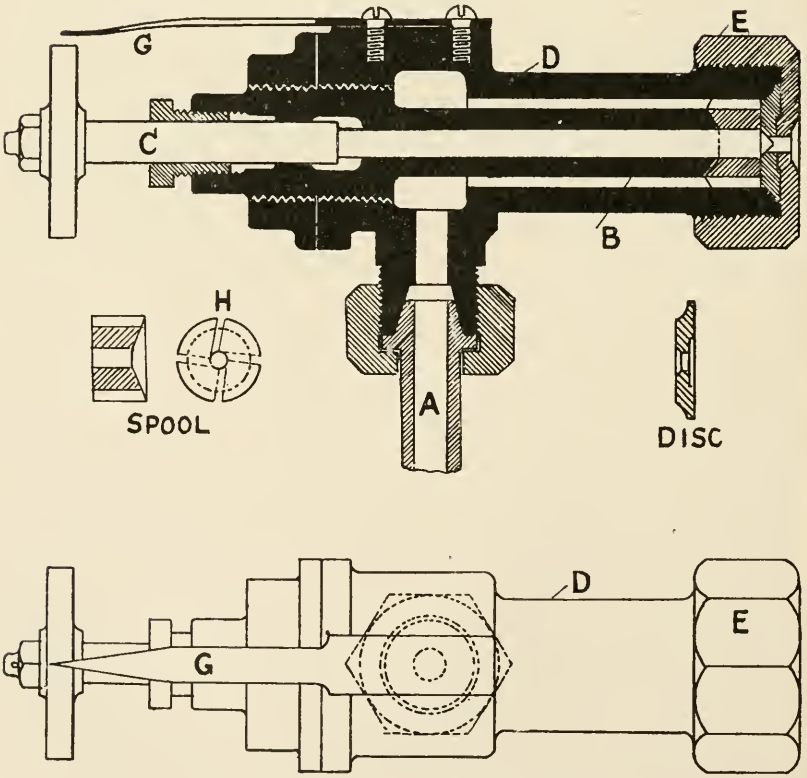


Fig. 99.—Kermode Oil Fuel Pressure Jet Sprayer.

through the nozzle. An indicator G shows the extent of the opening. Both the spool and the disc are removable, and can be fitted to suit any rate of burning within a fairly wide range. When renewed, the spool and disc make a practically new burner, and this system of renewal is a convenient one ; it is embodied in all the better types of burner.

Hot Compressed Air Burner.—In this burner the oil is partially vaporised and sprayed by means of hot air at a pressure of from $\frac{1}{2}$ lb. to 4 lb. according to the particular requirements. A section through the burner is shown in Fig. 100. The oil enters at A and its flow is regulated by a conical valve operated by the wheel E. The air heated by waste gases in the uptake or elsewhere enters the burner through B and C, of which the air entering through C mixes with the oil and vaporises it. To obtain a complete mixing a helix K is fitted in the

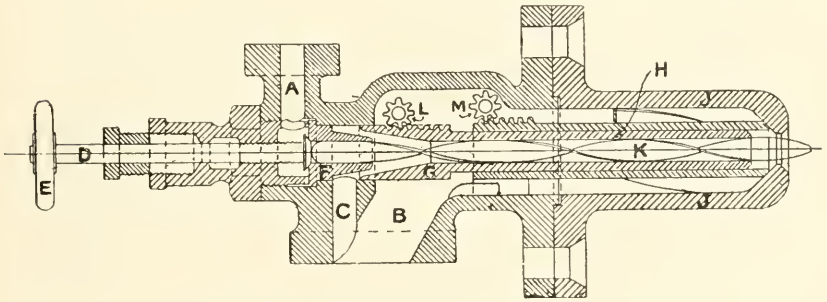


FIG. 100.—Kermode Hot-air Pressure Sprayer for Oil Fuel.

central tube G. The vaporising air supply is regulated by the rack and pinion L, and is entirely independent of the air supply for combustion, which is regulated by another rack and pinion M. All the elements of combustion are thus under complete control, and additional air for combustion is induced by the flame and draught from around the exterior of the burner for which provision is made in marine boilers.

The burners are started by opening the air supply at C to the full extent, and then starting the compressor to blow through the furnace, and ensure that no explosive vapours are present. This type of burner is much used for industrial purposes in which a good dry heat is required.

Oil Fuel Heater.—It has already been pointed out that when the oil is heavy it atomises very badly at a low temperature, but by heating it to a moderately high temperature the oil becomes thinner and, running freely, atomises easily. For high rates of combustion an oil

heater is necessary to good combustion, and one is shown in Fig. 101. The heater is of the horizontal surface type, and the oil circulates through the upper tubes and returns through the lower rows. Between the upper and lower rows of tubes a diaphragm plate is fitted, so that the steam when condensed generally fills the lower space, access to which is given through $\frac{1}{4}$-inch holes in the diaphragm plate, about 12 at the steam inlet end *A*, and about 2 at the outlet end *B*. A drain conveys away the condensed water as required. It is preferable for the oil to pass through the tubes internally, so that the surfaces can be more easily cleaned by a tube brush.

The collar bolts shown in Fig. 101 are frequently found to stretch and cause external leakage at the joint or, what is more serious, leakage of oil fuel into the steam space of the heater, and thus into the feed

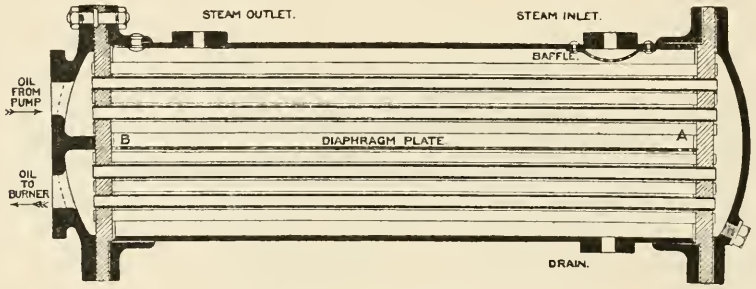


Fig. 101.—Oil Fuel Heater.

water system. The use of jointing material in this position being inadvisable, it is difficult to make them steam and oil tight under the high working and test pressures required, and this is particularly the case with the large heaters now in use.

In an improved design made by Messrs. Cammell-Laird, shown in Fig. 101A, the sandwich joint and collar bolts are eliminated by combining the tube-plate and distance piece or oil chamber into a solid steel forging, which is accurately machined to fit the body and riveted to it. The cover is then secured by plain studs and nuts. This form of construction has also been applied to Oil Fuel Heaters of the straight tube type as well as the U-tube type shown in the figure. The oil chambers are of sufficient length to allow the oil inlet and outlet branches to be formed on, or attached to them, and to afford the necessary facility for caulking the riveted joints.

The arrangement of the retarders should be noticed in the annexed illustration, the outlet allowing a freer flow than the inlet.

Oil Fuel Filter.—To prevent gritty matter choking the burner openings, which are very small, the oil is filtered after being heated

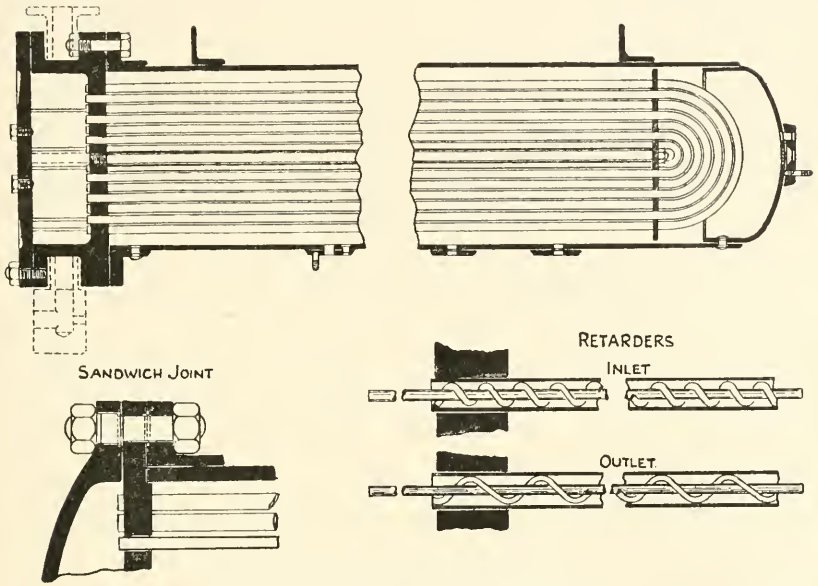


FIG. 101A.—Cammell-Laird Oil Fuel Heater.

and before it reaches the burners. Strainers of 36 and 24 meshes to the inch of wire gauze are fitted respectively in series on an outer and inner

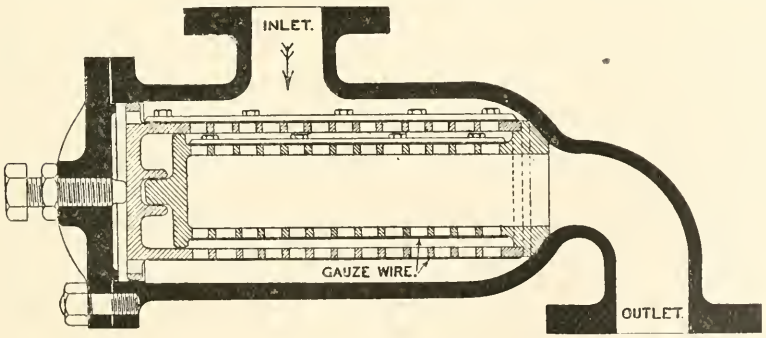


FIG. 102.—Oil Fuel Filter.

cylinder, as shown in Fig. 102, and can easily be removed and cleaned.

Water evaporated per 1 lb. of Fuel.—The combustion of 1 lb. of the best Welsh coal will, theoretically, produce about 15 lb. weight

of steam ; but in practice, only 9 to 12 lb. are produced. From many practical experiments with oil fuel it may be taken that from 14 to 18 lb. weight of steam can be produced by the combustion of 1 lb. of fuel oil. In practice 15 lb. is a good average.

The relative values of oil and coal, based on their practical evaporative power, are probably in the ratio of about 5 to 3.

The efficiency of evaporation depends on the rate of evaporation, and the rate of evaporation must depend on the rate of combustion, which affects both coal and oil fuels. The efficiency of combustion of oil fuel falls off less quickly than that of coal, because the furnaces and tubes usually remain clean, which constitutes an advantage.

Advantages of Oil Fuel over Coal.—Assuming the same weight to be carried in each case, the advantages of oil fuel may be considered as below :—

1. Greater energy available in the fuel ; increase about 30 per cent., or based on its evaporative capacity in practice an increase of over 60 per cent.

2. Extension of steaming distance ; increase about 60 per cent.

3. Reduction in space for stowage ; decrease about 10 per cent.

4. Ease of transhipment and stowage, probably at the rate of 300 to 400 tons per hour through a short length of pipe.

5. Reduction in number of firemen ; decrease about 80 per cent.

6. Maximum speed at short notice, with continued efficiency, because there are no fires to clean, but tubes must be swept at intervals.

7. Less wastage at varying powers, because the fuel-supply can be shut off at once from the furnaces, or varied to suit the power required. With coal, the furnaces cannot be cleared of the coal in them, which roughly amounts to about one ton for each boiler, and this amount is necessarily wasted when the ship is stopped.

8. Generally it is an economical advantage to fit out a vessel for oil fuel combustion when the supply is assured and the price of oil fuel is not greater than $2\frac{1}{2}$ times that of coal. All the advantages originally claimed by the author in his lectures to the R.N. War Colleges in 1903–6 have been more than realised during the war, and it might perhaps be claimed that this forecast gave an incentive to its use for naval and marine purposes generally.

Oil Fuel Burning in Practice.—The various fittings in connection with oil fuel burning have been described, with their uses, and the following, compiled from Messrs. Yarrow's instructions for their system, will be useful for guidance.

Taking oil fuel on board.—No liquid fuel should be put on board which has not been tested for flash-point, and if the flash-point is below 200° for shale oil, or 175° F. for other oils, it should not be accepted. The fuel should be filtered through large filter screens to keep it as clear as possible from dirt, etc.; the filters fitted in the boiler rooms are only intended to deal with small impurities in the oil.

The oil fuel tanks should be examined every day for an accumulation of water at the bottom. This water can be pumped out by a semi-rotary pump fitted with a suction pipe leading to the bottom of the tank, and delivering into a bucket.

No oil should be allowed to accumulate in the air boxes, bottoms of furnaces, bilges, or on the boiler-room floor plates, etc., and no lighted material should be allowed access to the bilges; after steaming, the bottoms of furnaces, etc., should be specially examined, and any oil at once removed.

If a leakage from the oil system should occur at any time, the oil supply should be immediately shut off by means of the stop valves and the pumps stopped.

When a fitting is opened out for examination or cleaning, oil-trays should be placed to catch the oil or liquid.

Raising steam.—The hand pump should always be ready, so that the boiler can continue to be supplied if the steam pump stops for any reason.

Before lighting up, and while everything is cold, great care should be taken that all burners are closed, the air boxes, furnaces, etc., clear of oil and well ventilated, and when lighting up or relighting a burner the operator should stand well clear of the sight holes.

The portable auxiliary oil fuel heater is used for lighting up, and some loose cotton waste, previously well soaked in oil, should be placed in the lamp tray and ignited with a torch after filling the body of the auxiliary heater with oil. The heater should be well heated up before the "lighting up" burner is turned on, and should on no account be left unattended when lighting up, as otherwise there is a great danger of flooding with oil.

When the oil is relatively cold it escapes without making any noise, so that there is not the usual indication of the amount passing, and at the same time the combustion is sluggish, and a quantity of oil may collect. Any collection of oil is dangerous, because the furnace gets hot and this oil may generate gas quickly or boil over into the boiler room.

The hand oil pump is used to force the oil through the system until steam is available to drive the oil fuel pumps.

Each burner should be fixed at its proper distance from the end of its air distributor, and so that its centre line is, as nearly as possible, central to the air distributor. The position at which the oil spray issues from the burner should be such that the air distributor is just filled with sufficient flame to keep the air distributor dry without becoming red hot. (Compare Kermode system.) If the oil spray leaves the burner at an incorrect position, or is not central with the air distributor, flaming in the air box, or dripping of oil from the air distributor may result.

Should a burner be extinguished the cause should be traced and may be due to, probably, any of the following :—

(*a*) Air passing over with the oil from the air vessel on the air pump.

(*b*) Water mixed with the oil, coming from fuel tanks or from leaky heater.

(*c*) Solid matter choking the burner, either through the filters not removing foreign matter, dirt, etc., or to carbonising of the oil.

(*d*) Too high an oil temperature.

(*e*) Excessive air supply through the air distributor, and the air doors should be shut for a few seconds.

When a burner is choked and cannot be cleared by temporary alteration in the spindle adjustment, it should be at once removed and thoroughly cleaned. Cleaning should be very carefully carried out so that the outlet holes are not roughened, enlarged, or altered in shape. Burners should be taken to pieces and cleaned periodically, and should never be left in place when disconnected.

On easing down or stopping, the oil supply should be reduced before the air supply, so that any inflammable gas may be blown out.

The oil pressure depends upon the rate of combustion required in connection with the number of burners used, but usually the pressure should not exceed 180 lb. per square inch at the burners.

Under normal conditions, if the oil pressure exceeds 150 lb. it is generally necessary to increase the number of burners, but if the pressure falls below 75 lb. it is necessary to close down some of the burners.

The required oil pressure should be kept as constant as possible. When it is unsteady special attention should be paid to the oil levels in the air vessels attached to the oil fuel pumps, which should be kept low, but not too low to show in the gauge glass.

When the rate of steaming varies, the corresponding change in the

oil supply should generally be made by altering the oil pressure, and not by readjusting the amount of opening of the burners, unless readjustment is necessary to maintain a fine enough and satisfactory oil spray. In some cases it will be necessary to shut off some of the burners at the distributing valve boxes.

The oil is raised to a suitable *temperature* to make it fluid enough for efficient spraying, and its temperature at the burners under working conditions is dependent upon the character, viscosity, and class of oil to be burned, and varies from a maximum of about 220° F. for the heavier oils to about 80° for Scotch shale fuel oil. Too high a temperature causes pulsation or unsteady burning in the furnaces. Too low a temperature may produce smoke or flaming in the uptakes, and may also injure the firebricks.

Before the steam is admitted to any oil heater, the valves between the heater and the oil fuel pump should be opened, and they should remain open until after the steam is shut off on ceasing to use fuel oil.

Under normal working conditions water should flow to the feed filter tank from the drain collector in connection with each oil fuel heater ; if, however, leakage of oil is observed in a drain collector, the mixture of oil and water should be drained off into buckets until such time as the standby heater can be brought into use. The drain valves on the drain collectors should be so regulated that a few inches of water is always showing in the water-gauges on the collectors.

The air supply doors and slides should be frequently worked to ensure that they are efficient ; any blocking of the air distributors may cause them to be overheated, flame to appear in the air boxes, dripping of oil, and excessive smoke.

Before lighting up, the corresponding air doors and slides should be opened and particular attention paid to the automatic doors and slides to ensure that no undue friction or other obstruction will prevent them from closing under influence of a back draught.

The smoke observation windows and mirrors should be cleaned at least once every 24 hours, and frequently used for the detection of smoke. When watching for smoke, care should be taken that the light fitted for the purpose is not confused with any other.

As soon as steam is raised the fans should be started, and they should not be stopped again while the burners are working. If the fans stop for any reason the oil supply to the burners should be shut off at once.

It is good practice to have all the burners at work in one boiler room open the same amount, so that if the proportion of air is correct for one burner it is about correct for all the burners, and the adjustment of air doors for all burners should be approximately similar.

When a set of burners is shut off, the air doors which supply the air to these burners should be closed also.

The quantity of *air required* depends upon the quantity of liquid fuel to be burnt, and must be regulated so as to produce :—

(1) An amount of smoke which is just visible at the funnel ; and

(2) A flame of about the same size from each burner of the same designed maximum output, or pattern.

Too small a quantity of air will produce excessive black smoke and tend to produce overheating of the air distributors. Too large a quantity will produce white smoke, and will lessen the efficiency of the boiler.

Under ordinary conditions, the correct quantity of air required for variations in oil consumption (and power) is best obtained by regulating the air pressure in the boiler room ; with the air doors open as wide as possible consistent with an efficient flame being maintained from each burner. The flame should, however, be carefully watched as some readjustment of the air openings may be required.

Deposits of carbon arising from the decomposition of the oil, depending on the quality of the oil and the care of the boiler room staff, are continually formed on the interior of the air distributors, and these should be removed by suitable cleaning tools before the deposit accumulates to such an extent as to cause trouble. The rate at which the carbon is deposited will be increased if the air is insufficient. Air distributors may require clearing at intervals of 20 to 30 minutes.

Oil dripping from the air distributors should be immediately attended to and is probably due to :—

(1) Dirty air distributors.
(2) Low oil pressure.
(3) Foul burners.

The oil filters and strainers should be examined to ensure that they are in proper order, and the pressure gauges fitted on each side of the filters should be frequently noted for indications of gagging or clogging. Under ordinary circumstances one filter of a pair is sufficient to pass all the oil required, the other being considered as a spare to be brought into use if necessary. Drain cocks are fitted to the bottoms of the

filters, and the cock on the filter to be cleaned should be opened before the filter is removed, so as to release any accumulation of pressure.

To shut down when using oil fuel, shut off steam from the pumps, thus allowing the burners to use part of the oil remaining in the system and to gradually burn out.

When oil fuel fittings are not in use, great care should be taken to keep them clean and in good working order.

PART IV

STEAM—THE WORKING SUBSTANCE

CHAPTER XIII

THE INDICATOR AND INDICATOR DIAGRAMS

The indicator is used to obtain a diagram showing the pressure at each instant of the stroke of the piston in a steam, or other fluid, cylinder; and from this diagram much other useful information is obtained. By the proper use of the indicator the mean pressure on each side of the piston can be obtained, and, when the other details are known, the indicated horse-power developed in the cylinder can be calculated.

The action is similar in nearly all indicators; but the motion of the pencil, used to mark the pressure on the diagram paper, is obtained by many different applications of the principle of parallel motion, or pantograph.

"Dobbie-M'Innes" Indicators.—There is a light cylindrical metal barrel, *O*, round which a paper is stretched, as shown in Fig. 105. The barrel is moved round on its axis by a string, *L*, which is kept in tension by a spiral spring inside the barrel. The other end of the string is attached to a system of levers, worked from a pin rigidly connected with the engine piston rod; so that in each stroke of the engine piston the barrel *O* is moved round a certain distance by the string, and on the return stroke of the engine piston it is returned by the spring to its original position. At the end of one revolution, that is, one forward and one return stroke of the engine piston, the string again pulls the barrel round, and so on, alternately.

The pencil *P* has no horizontal motion; the paper is moved past it by the movement of the barrel. The amount of traverse in relation

to any fixed point bears an exact ratio to the distance of the engine piston from a similarly situated point in its stroke.

There is another cylinder, *B* (Fig. 103), in which a piston, *A*, works,

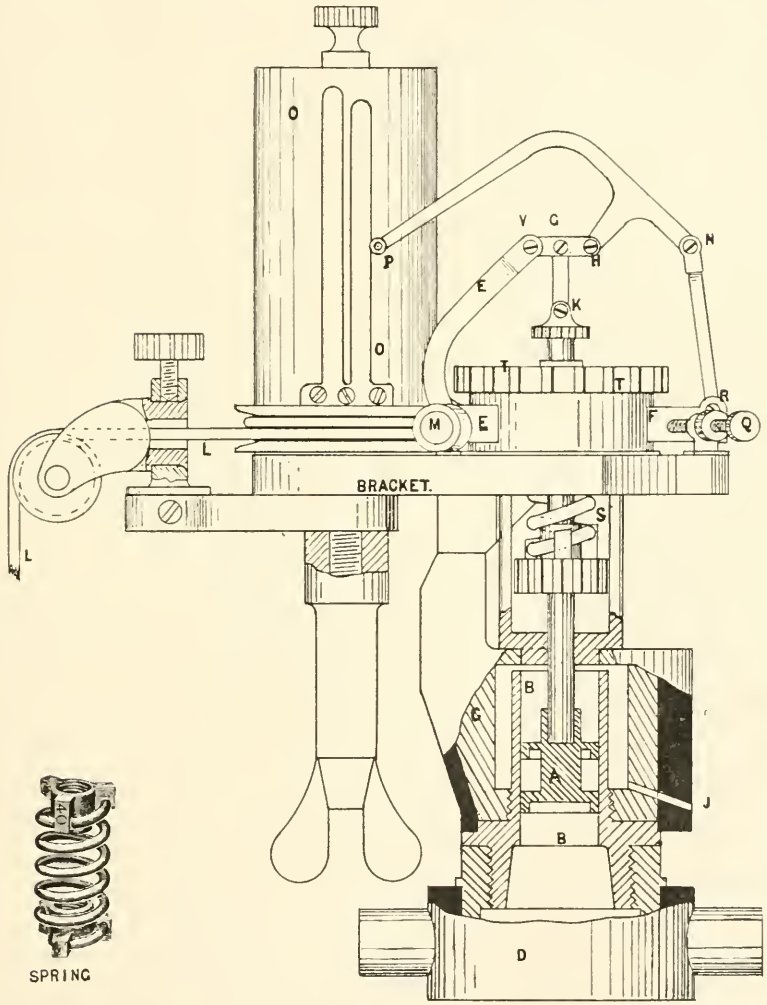


FIG. 103.—"Dobbie-M'Innes" Indicator (Design No. 1).

with its axis parallel to that of the cylindrical barrel *O*. The upper side of the piston, *A*, is open to the atmosphere through a small hole, *J*, drilled in the lower part of the cylinder *C*. The lower side of the

piston is connected by pipes with the ends of the engine cylinder, and through a cock it can be connected with either end as required. The piston is made of certain area, and loaded by a correspondingly proportioned spring, *S*, so that when the spring is compressed or extended a certain distance, the vertical position of the pencil represents a corresponding pressure *per square inch*.

For various cylinder pressures per square inch various springs are used, corresponding to various scales of compression, namely, 160, 120, 100, 80, 56, 40, 32, 24, 16, and 8 lb. per inch as measured, or marked, on the diagram. Thus, if a spring marked 40 lb. be in use, it means that a vertical movement of one inch of the pencil *P*, which is attached to the piston *A*, represents a difference of 40 lb. pressure per square inch on the engine piston.

The pencil *P* is attached, by a parallel motion, *HGK*, and a rod (shown in the figure), to the piston *A*. When the pressure below the indicator piston *A* is greater than that of the atmosphere, which tends to press the piston downwards, the piston rises correspondingly against the pressure exerted by the spring; but if the pressure below the piston is less than atmospheric pressure, the piston falls correspondingly with the lower pressure in the end of the engine cylinder, with which the indicator cylinder is in open connection.

EF is pivoted on the barrel *B*, so that the pencil *P* can be pressed on, or removed from, the paper at pleasure. A thumb-piece is fitted at *M* for this purpose, and by means of a screw, *Q*, and a stop, fitted on the bracket connecting *O* and *B*, the pencil *P* can be adjusted so as to press lightly on the paper.

Care should be taken, when selecting a spring, that the requisite pressure, both highest and lowest, can be registered on the paper.

Fig. 104 shows the exterior of a Dobbie-M'Innes indicator with the spring enclosed in the cylinder. This type is frequently used for ordinary marine and land type engines; but the external spring type (Figs. 103 and 105) is preferable, because the spring on which the correct indication of the pressure in the engine cylinder principally depends is not subject to large changes of temperature.

Fig. 105 is very similar in general design to Fig. 104, but a continuous roll of paper is contained within the drum *O* and protected from moisture by a cap on top of the drum. A detent gear is provided for stopping the motion of the drum without disconnecting the cord. The drum spring is fitted at *W* below the drum, and its tension can be varied at any time.

Theoretically the motion obtained with the present Dobbie-M'Innes indicator is exactly the same as in the M'Innes type (see 2nd edition), and the peculiar shape of the lever *PHN* in the pantograph motion is merely adopted to obtain a more uniform and direct motion from the piston, with less liability to frictional error. This motion multiplies the piston travel six times at the pencil point. In Fig. 103, a recent improvement is shown by which the cylinder cap can be removed at the same time as the cap *T* and the spring changed more quickly. The piston rod is made of steel tube to secure lightness and strength. The piston *A* is made of case-hardened steel, and the recess shown

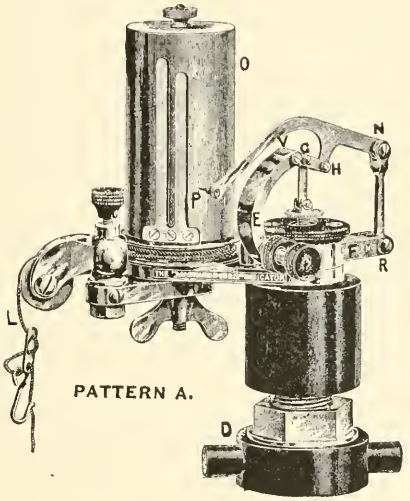


Fig. 104.—Enclosed Spring Indicator.

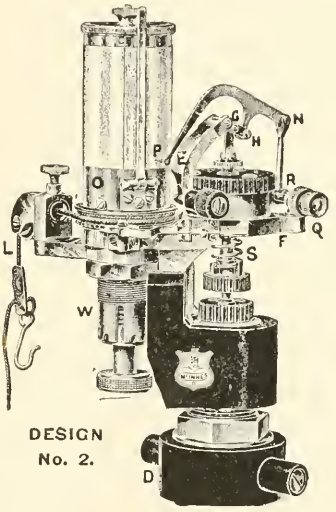


Fig. 105.—External Spring Indicator.

receives lubrication and grit. These indicators are generally made in three sizes: large, for ordinary steam and ammonia engines; small, for high speed steam engines; and half-size, for motor cars, launches, etc., and are also made in several designs to suit the special requirements of steam, Diesel and internal combustion engines.

The exteriors of the barrel *C*, the cover *T*, the thumb-screw, and the connecting nut *D*, are covered with vulcanised non-conducting material, so that the indicator can be easily handled when warm. To obtain a true diagram, it is necessary to previously thoroughly warm the indicator cylinder and connecting pipes. A small drain-hole is fitted to the indicator cock, immediately below *D*, by which the condensed water can be blown out, and it is advisable to allow the

piston and pencil to work freely for a short time before use, by connecting up each end of the cylinder alternately.

By setting the indicator cock so as to connect the drain-hole with the space below the piston *A*, the atmosphere has free access below the piston, and as it always has free access above *A*, through the holes previously mentioned, the piston can thus be held in a position due to the atmospheric pressure only. The line marked by the pencil *P*, when *A* is so governed, is called the atmospheric line (*GC*, in Figs. 106 and 107). The absolute zero line *OA*, from which the pressures are measured, is drawn horizontally, to scale, 14·7 lb. below this atmospheric line.

The horizontal length of the line made by the pencil may be taken as representing the length of the stroke of the engine piston.

The vertical height of the pencil *P* and the mark made by it at any point above the absolute zero line represents the pressure absolute per square inch on the engine piston at that point, and its position measured horizontally (*i.e.* parallel to the atmospheric and zero lines) represents the relative position of the engine piston in its stroke.

The diagram shows, for one revolution of the engine, the pressure in the cylinder on one side of the piston only. By connecting the indicator with the other end of the cylinder, another figure is obtained, showing the pressure, for one revolution of the engine, on the opposite side of the piston; and from the pair of diagrams the difference of pressure on the piston at any selected point can be measured. For a perfect representation of the pressures in the cylinder at the same instant on opposite sides of the piston two indicators are necessary, but for practical purposes it is quite sufficient to take the diagrams within a few seconds of each other on the same card or paper.

Elementary Diagram.—Fig. 106 shows an elementary diagram which might be obtained from an engine cylinder under instantaneous conditions of admission and cut-off.

GC represents the atmospheric line;
OA ,, zero absolute-pressure line, 14·7 lb. below *GC*;
OF ,, steam initial-pressure line measured vertically;
FE ,, steam line, duration of admission;
ED ,, expansion line, after steam is cut off;
DA ,, final pressure, after expansion;
BP ,, back-pressure line, cylinder open to exhaust during nearly the whole of the stroke.

Ideal Diagram.—If the expansion is carried to its utmost extent, B and D will coincide; but this extreme ratio is never reached in practice because the forward pressure should always be sufficiently in excess of the back pressure to overcome the initial resistance of the engine. In some specimens of actual diagrams the points actually coincide, but this is principally from cylinder condensation, producing a lower pressure than would otherwise exist.

Actual Diagram.—Fig. 107 shows a diagram obtained in actual

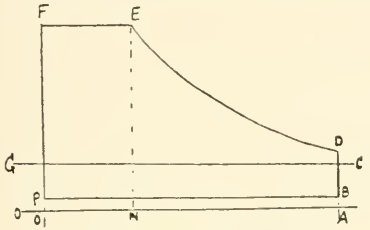


FIG. 106.—Elementary Diagram.

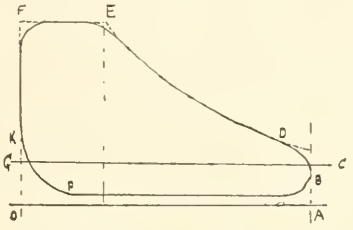


FIG. 107.—Actual Diagram.

practice, which does not differ much from the elementary diagram shown in Fig. 106. All the points are rounded off. At F the full pressure is not obtained until a little after the beginning of the stroke, because the steam is admitted gradually while the valve is uncovering the steam port; for a similar reason, the cut-off is gradual, and the point E is rounded off because the pressure falls gradually. The expansion line remains about the same until the point D, where exhaust commences, is reached, when the pressure falls gradually until the end of the stroke. The exhaust is not wide open until after the beginning of the return stroke; this produces the rounding of the curve at B. The exhaust line is horizontal until P is reached, when the exhaust is gradually closed, while the steam in the cylinder is compressed behind the piston and the pressure rises until, just before the end of the return stroke, admission commences, when the pressure rises still further, and frequently attains the initial pressure ready for the next forward stroke.

The rounding at all these points affects the working of the engine, generally in a favourable manner. Thus, at D there is no sudden fall of pressure when the piston is changing its direction of motion. From P to K the pressure rises gradually, and not suddenly, the piston is brought slowly to rest without shock, and the full pressure is reached at about the time it is required for the forward stroke. This rounding at PK is called the *cushion or compression corner*. The steam and back

pressure lines should generally be horizontal, and the admission and final pressure lines should generally be vertical; but in practice, as will be shown later, the figures vary considerably, and each difference is due either to some peculiarity of the engine or its working. These are not necessarily either faults of construction or of working, and are frequently the outcome of economical experience.

Errors due to the Indicator.—Unless the indicator is kept in good order and perfectly clean, the piston, pencil, and other sliding parts have a great tendency to produce jerky movements. *Friction* also reduces the mean pressure recorded, by raising the back pressure, and by decreasing the height of the forward pressure.

The variations in the pair of diagrams shown in Fig. 108 have been produced by friction, probably between the indicator piston and its cylinder, from gritty matter or lack of proper lubrication. Instead

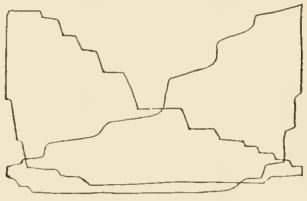


Fig. 108.—Friction of the Indicator.

Fig. 109.—Undulation of the Indicator.

of a gradual curve there are a series of steps, formed by undue frictional resistance holding the piston until the difference of pressure overcomes it and produces some small movement. The example shown is a very bad case; in practice it is generally less accentuated, and more nearly approaches that due to the undulation of the indicator.

Undulation or vibration of the indicator is shown in Fig. 109, and is produced from various causes—either due to the indicator not being suitable to the number of revolutions of the engine, or to the spring not being suitable to the pressure. The example was taken with an old pattern Richard's indicator, and that shown in Fig. 119 was taken from the same engine, at about the same power and number of revolutions, with a later pattern indicator. The substitution of a stronger spring sometimes overcomes the effect of undulation by allowing less travel of the indicator piston; and by shortening the travel of the paper past the pencil, horizontally, some improvement can also be made. The working length of one of the levers which move the

string can generally be altered by removing one of the pins in the lever connections and fitting it into another hole provided for the purpose.

If the indicator piston is moved at great velocity by a very quick reciprocation of the engine, and the weight of the moving parts of the indicator is unnecessarily great, it increases the tendency which the pencil always possesses to rise too high and fall too low. The frictional resistance of the piston in the cylinder, the pencil on the paper, and in the joints of the mechanism, tend to check this excessive movement of the pencil; but when the pencil arrives at the upper and lower extremities of its travel, the frictional resistance tends to keep the pencil either above its normal position when falling, or below its normal position when rising, and therefore the area of the figure contained within its boundary lines may be slightly larger than the perfect figure, obtainable if these defects did not exist. Other errors possibly occur from inaccurate fitting of the joints in the mechanism—inaccurate calibration of the spring, when equal degrees of compression are not exactly to scale; and variation in the length of the string (or the virtual length, if pin or sliding joints are used in connection with it), caused by difference in the tension exerted by the spring.

All these errors are nearly avoidable, and, although they should never be overlooked, can generally be neglected for the practical purposes for which the indicator is used.

The principal error affecting diagrams in actual practice is caused by wiredrawing in the pipes connecting the indicator cylinder with the engine cylinder, and for naval purposes the pipes have recently been increased in area to obtain more accurate reading. The velocity of the steam through these pipes should not as a rule exceed 6000 feet per minute, from which the area can be calculated.

Faults shown by the Diagram.—There are in practice many faults which may be gathered from a study of the diagrams taken from a cylinder, as well as the useful information relating to the horse-power, steam consumption, and other matters of calculation. The faults are given in detail as follows:

Length of the String.—If the string be not the correct length, the barrel of the indicator comes to rest before the end of the piston stroke, and therefore it is necessary to test this adjustment by taking a trial diagram. The effect is shown by the general appearance of the admission and final pressure lines, one of which will probably be vertical, with square corners both at top and bottom.

Wiredrawing.—If there is insufficient opening of either the steam or the exhaust ports, wiredrawing occurs, but care should be taken not to confound this result with that obtained by using an early cut-off when "linked up."

If the opening to steam is very much restricted, the steam pressure is reduced before the point of cut-off is reached, producing a gradual fall of pressure on the admission line before expansion commences. This is shown in Fig. 110, where the highest pressure is not reached until a little after the beginning of the stroke, and at the point of cut-off it has fallen to L instead of remaining at E. The absolute pressure at this point is represented by the height NL instead of NE.

Restricting the opening to exhaust is a very serious loss, and the effect is shown in Fig. 111. The consequent increase of back pressure reduces the mean effective pressure on the piston, and thus reduces

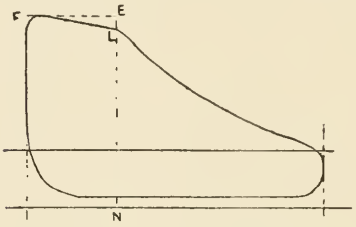


FIG. 110.—Steam wiredrawn.

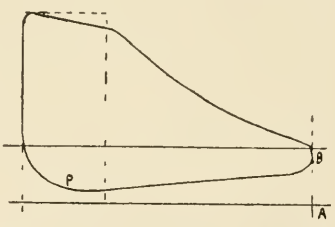


FIG. 111.—Steam and Exhaust wiredrawn.

the power indicated by the engine. The defect is a common one on early trial trips, but is generally remedied after discovery by enlarging the exhaust ports both in the slide valve and in the cylinder false face. To obtain an unrestricted exhaust, negative exhaust lap is frequently used in fast-running engines.

In Fig. 111 the result of wiredrawing both the steam and exhaust is shown. The exhaust pressure is not fully released at the beginning of the return stroke, as shown by BP, which is not horizontal.

Leaky Slide Valve.—If the slide valve is leaky, the steam continues to enter the cylinder after cut-off has taken place, and instead of the expansion line being as shown in dotted lines in Fig. 112, there is a continuing higher pressure, as shown by the full line EQB.

Some steam leaks past the steam edge of the valve, without entering the cylinder, into the exhaust side, and raises the back pressure, as shown by BRS instead of BPK, thus reducing the power obtained in the cylinder with a greater expenditure of steam. The loss with fast-running engines is considerably less than with slow-running engines,

where the steam has plenty of time to leak past the edges of the valve. In stage-expansion engines a small percentage of leakage is not so important in the high and middle pressure cylinders, because the leakage is eventually used in the low-pressure cylinder, and not entirely wasted. The flat valve is supposed to be less liable to leakage than the cylindrical slide valve, and therefore flat valves are frequently fitted to low-pressure cylinders. It is especially important that no leakage should occur through the low-pressure slide valve, because, in addition to the actual leakage and consequent waste of steam, there is a tendency to increase the back pressure by decreasing the vacuum in the condenser (as shown in the figure, which is somewhat exaggerated).

Note.—In all diagrams the front end of the cylinder is next the crank.

Leaky Piston.—A pair of diagrams are shown in Fig. 113 which

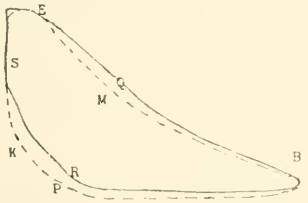


FIG. 112.—Leaky Slide Valve.

FIG. 113.—Leaky Piston Packing.

were taken from the high-pressure cylinder of a compound engine. The right-hand diagram, taken from the back end of the cylinder, varies but little from that obtained in ordinary practice, when properly "linked up" for a low power.

The diagram taken from the front end is abnormal, showing large fluctuation of back pressure, with an average above the normal. Pressure has apparently been allowed to pass from back to front, but not from front to back, as shown by the front-end diagram; leakage is also evident from back to front, as shown by the back-end diagram. The back-end diagram shows an almost immediate fall of pressure at the beginning of the stroke, and there is a corresponding rise on the opposite side of the piston. This points to leakage from back to front. The front-end diagram shows a normal fall of pressure from the beginning to the end of the stroke; but on the return stroke the exhaust or back pressure is too high, and confirms the leakage from back to front. Such leakage could be caused by either the packing ring not fitting properly between the junk boring and the dy of the piston, or by the

tongue piece not properly covering the opening between the ends of the packing ring. The piston, when pushing the packing ring and tongue piece before it, might be expected to maintain a steam-tight joint between it and them ; but when the piston is dragging the ring and tongue, some leakage might be expected either by way of the tongue or through the space left between the junk ring and the body of the piston, and thence through the space between the packing ring and the flange. In this instance the packing ring was found a little slack between the junk ring and the flange of the piston, and the tongue was in a similar condition. A thin liner was removed from between the junk ring and the piston, no alteration was found necessary in the tongue, and subsequent diagrams were of a normal character.

Such a leak is difficult to correct, as it is always possible to nip the packing rings too closely. Accurate fitting can only be obtained by removing the piston from the cylinder, which may take a long time, as frequently the piston cannot be withdrawn from the rod without considerable trouble and delay. When the piston is replaced on the rod plenty of dry black-lead should be smeared on the joining surfaces, and little trouble is then experienced in subsequent removal.

Cylinder worn unevenly.—In Fig. 114 some diagrams, taken some years ago from a L.P. cylinder in H.M.S. *Colossus*, are shown. The cylinder barrel was worn in such a way that there was considerable variation of the actual diameter in which the piston worked. The packing ring therefore continually varied the opening at the tongue piece or sliding joint, and allowed varying leakage through this opening. The diagrams were taken at very low pressure and power, the initial pressure in the cylinder being only 5 to 6 lb. absolute, with a vacuum of about 28 inches in the condenser. The points to notice are : that up to the point of cut-off the forward pressure is nearly maintained, because the leakage is not great enough to appreciably lower the pressure ; towards the middle of the stroke, after admission ends, the pressures on the opposite sides of the piston rapidly equalise, and for the latter part of the stroke there is no difference of pressure, because all the steam has taken a short cut to the condenser ; the opening to the condenser is open nearly the whole of the stroke, and consequently the exhaust line is almost horizontal throughout the stroke ; the cushion-

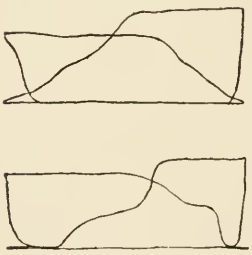


Fig. 114.—Cylinder worn.

ing corners are peculiar—at one end no compression is apparent, and is probably prevented by leakage, while at the other end some cushioning is apparent, indicating less leakage, and that the working barrel is less worn at this part.

Angular Advance of the Eccentric too small.—In this case all the operations of the valve, including admission, cut-off, release, and compression, are late; and instead of the diagram shown in full lines in Fig. 115, one similar to that shown in dotted lines results. This diagram shows similar characteristics for both the forward and the return strokes; whilst an alteration in the length of the rod produces characteristics of opposite kind for forward and return strokes, as exemplified in Figs. 117 and 119.

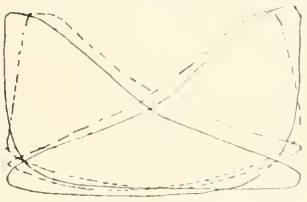


FIG. 115.—Angular Advance too small.

Angular Advance of the Eccentric too great.—In this case all the operations of the valve are too early, and a diagram such as shown in Fig. 117 results, with perhaps steam and exhaust loops. The characteristics obtained by advancing the eccentrics are similar for both the forward and return strokes.

The effect of *linking up*, or shortening the link, produces a similar result, by reducing the travel of the slide valve on the false face.

After an engine has been designed and completed, the angular advance is sometimes altered to produce a more accurate result in the power developed, or to suit a difference in the pitch of the screw. If the angular advance is decreased, a later cut-off is obtained, and a greater power can be obtained from the engine, while an increase produces an earlier cut-off, and allows less power to be developed. It does not follow that either operation will produce a difference in the economy of steam, but within moderate limits, as pointed out elsewhere, an early cut-off produces economy.

Slide Valve incorrectly set on the Rod.—In this case the slide valve has too much *steam lap* at one end and not sufficient at the other, while the exhaust lap is insufficient at the first end and too great at the latter. Diagrams taken in actual practice are shown in Figs. 116 to 119.

In Fig. 116 the rod is much too long. There is no semblance of equality in either initial or mean pressures, and the steam admitted at the back end is considerably over-expanded; while probably an

exhaust loop would have been formed, if the heat of the cylinder walls had not partially re-evaporated the condensation water, and thus produced steam of a slightly higher pressure than the back pressure on the piston during the return stroke. Although steam is cut off much later at the front end, there is a rapid diminution of pressure, because the heat, abstracted during the previous stroke, has to be replaced in the cylinder walls. The mean cut-off, as nearly as could be measured, was at about 30 per cent of the stroke, and the diagram was taken at a comparatively low power.

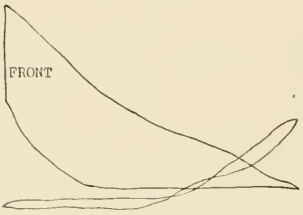


Fig. 116.—Rod much too long.

In Fig. 117 the rod has been shortened, but is still too long. This is shown by the inequality of the mean and initial pressures, the cut-off being too early at the back end and too late at the front end. The too early cut-off at the back produces over-expansion, which is shown by an exhaust loop.

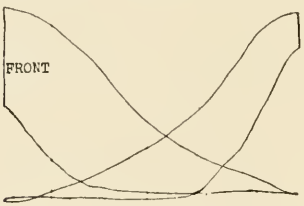


Fig. 117.—Rod too long (moderate power).

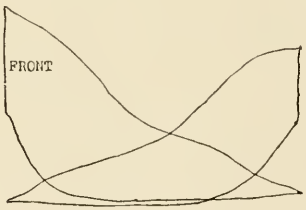


Fig. 118.—Rod too long (high power).

Fig. 118 shows a pair of diagrams taken at the same slide-valve setting and cut-off, about 33 per cent, but at a higher power and speed.

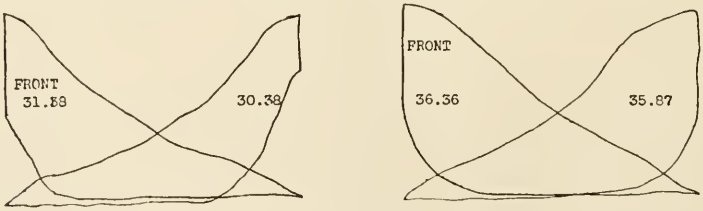


Fig. 119.—Rod about correct length.

The same characteristic features are shown as in Fig. 117, but there is no exhaust loop ; this may be due to the slight difference in the cut-

off, or to less cylinder condensation, with a higher rate of revolution and a higher power.

Fig. 119 shows diagrams taken after shortening the rod still more, and the mean and initial pressures are now nearly equal. The exhaust loop is not evident, and for ordinary working, with the link considerably shortened to cut off at 30 per cent, instead of 60 per cent of the stroke when working at full power, the diagrams are satisfactory.

Clearance.—The effect of clearance is to vary the mean pressure obtained in the cylinder. The piston cannot be allowed to touch the cylinder cover or end at each end of the stroke, and consequently a small amount of space is allowed for safety. When the engine bearings wear, and so alter the exact position of the piston in the cylinder at the ends of the stroke, it tends to alter the virtual length of the rods, generally shortening them, and thus brings the piston downwards in a vertical engine. The steam passages between the slide valve and cylinder must also be considered. The sum of these several clearance volumes is ascertained, and an equivalent calculated in a fraction of the length of the stroke. The cut-off takes place also at a fraction of the stroke. The sum of these fractions, *less* the fractional amount of steam left in the cylinder at the end of the stroke, actually represents the volume of steam admitted at each stroke. The effect of clearance is shown graphically in Fig. 120, where the expansion line *ED* is somewhat higher than *EB* in the elementary diagram in which clearance has not been considered. Clearance affects the rate of expansion, for, in addition to the initial volume *FE* of steam admitted to the cylinder between the beginning of the stroke and the cut-off, an extra amount *FF′* is admitted to fill the clearance at each stroke.

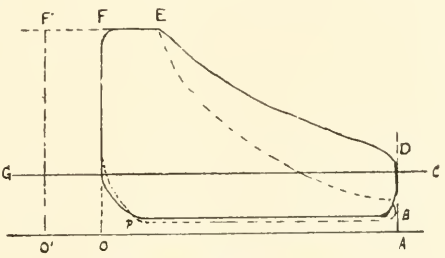


FIG. 120.—Effect of Clearance.

The actual amount is best obtained from the calculation of the weight of steam used per stroke, which is explained later. For the present calculation, the amount of steam *left* in the cylinder at the end of each stroke is neglected.

Let $\frac{1}{r}$ represent *FE*, the fraction of the stroke at which steam is

cut off; and let $\frac{1}{c}$ represent FF', the clearance, as a fraction of the stroke-volume.

$$\left.\begin{matrix}\text{Then the volume of steam}\\ \text{admitted at each stroke}\end{matrix}\right\} = \frac{1}{r} + \frac{1}{c} = \frac{r+c}{rc}.$$

At each stroke this volume of steam is expanded into the volume of the cylinder struck out by the piston, added to the clearance, that is, into a volume represented by $1 + \frac{1}{c}$.

$$\text{Therefore the } \textit{actual} \text{ ratio of expansion} = \frac{1+\frac{1}{c}}{\frac{r+c}{rc}} = r\left(\frac{1+c}{r+c}\right).$$

For example, take a cylinder in which the cut-off, as shown by the slide valve (or *apparent* cut-off), is at one-fourth stroke, and suppose the clearance is equal to one-eighth the stroke-volume of the cylinder—

$$\text{Then the } \textit{actual} \text{ ratio of expansion} = 4 \times \frac{1+8}{4+8} = 4 \times \tfrac{9}{12} = 3.$$

So that instead of obtaining *four* as a ratio of expansion, the actual ratio is reduced to *three*, from the effect of clearance.

The waste of steam resulting from clearance is reduced by the compression at the end of the return stroke. When the pressure is raised by compression in the clearance spaces, just before the point of admission, to the initial pressure, the loss from clearance is very small; but as this increases the mean back pressure, there is a decreased effective mean pressure. In simple or single-stage expansion engines it is not economical to compress the steam to this extent; but for higher stage expansion engines, when running at low powers, the economy of steam can be greatly improved by "linking-up," until the pressure of the cushioning steam in the H.P. cylinder is nearly equal to the initial pressure, thus producing the highest practicable total rate of expansion. The separate link is very useful for this purpose.

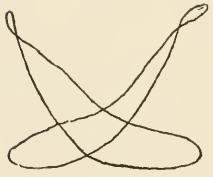


Fig. 121.—Steam Loop.

Steam Loop. — When the pressure of the cushioning steam exceeds that of the initial steam, it shows itself by a loop, as seen in Fig. 121. It should not in practice be allowed to quite attain this pressure, but something just below it.

In calculating a diagram with a loop, the area of the loop should be deducted from the area of the diagram, because it represents work done against the direction of motion of the piston, and thus reduces the *mechanical efficiency* of the engine.

Exhaust Loop.—On reference to Fig. 117, a small loop is seen in the left-hand corner, at the tail end of the diagram, taken from the back end of the cylinder. This is due to the pressure in the cylinder falling below that of the ordinary expansion curve, which is nearly adiabatic, and is the result of either—

1. Too high initial pressure, combined with too early cut-off, generally obtained by linking up too much ; or

2. Steam jacketing being insufficient to prevent undue cylinder condensation.

3. Incorrect setting of the slide valve.

Exhaust loop seldom occurs in the L.P. cylinder, because the range of temperature in the cylinder itself is frequently less than in the other cylinders, and the cut-off is possibly later ; both of which are reasons for less apparent condensation in the L.P. cylinder.

In the H.P. cylinder the initial pressure may be unnecessarily great for the power developed by the engine. For naval service it is usual to keep an otherwise unnecessarily high pressure for possible manœuvring purposes, and for water-tube boilers a minimum pressure of 140 lb. is recommended. A moderately high pressure is sometimes more economical for auxiliary purposes.

The economical advantages of high expansion cease when exhaust loop occurs, so far as the cylinder itself is concerned, but it is still admissible if it does not affect the smooth working of the engine, and if the power developed in the cylinder is not below that developed in another cylinder of the same engine. In some instances a judicious use of the cylinder jacket may cause the loop to disappear.

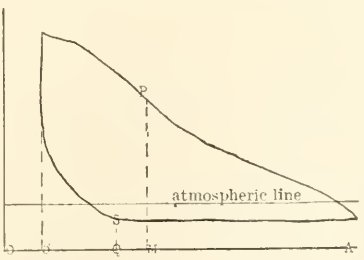


FIG. 122.—Weight of Steam used.

Measurement of the Weight of Steam used, from the indicator diagram.—In Fig. 122, OA represents the zero line of absolute pressure ; $O'A$, the length of the diagram, represents the stroke-volume ; and OO' represents the clearance on the same scale. Take a point, P, in the expansion curve after cut-off has taken place and where the steam

contains least moisture, and drop a perpendicular, PM, to OA. Then, ($OM \times$ area of piston) represents the volume of steam apparently admitted to the cylinder during the stroke, and PM represents the absolute pressure. The weight of a cubic foot of steam at a pressure, PM, can be found in the table at the end of Chapter II., and the weight of steam apparently admitted at each stroke can then be calculated.

Next, take a point, S, in the exhaust curve just after cushioning or compression has commenced, and drop a perpendicular, SQ, on OA; then, ($OQ \times$ area of piston) represents the volume of steam which does not escape from the cylinder at each stroke, and SQ represents its absolute pressure. The weight of a cubic foot of steam at a pressure, SQ, can be found on page 18, and the weight of steam which does not escape at each stroke can be calculated.

If it be assumed that the steam, at the points of measurement, is dry saturated, then the *difference* between the weight of steam apparently admitted during the stroke and the weight of steam which does not escape is the actual weight of steam which is used at each stroke. If w represents this amount, then the weight of steam used per I.H.P. per hour is equal to $w \times$ number of strokes per hour divided by the I.H.P.

In practice this amount is always below the actual amount, and it can never be in excess unless the steam is superheated considerably, and even then the result is not usually great enough, because the steam generally contains moisture which is not perceptible on the diagram.

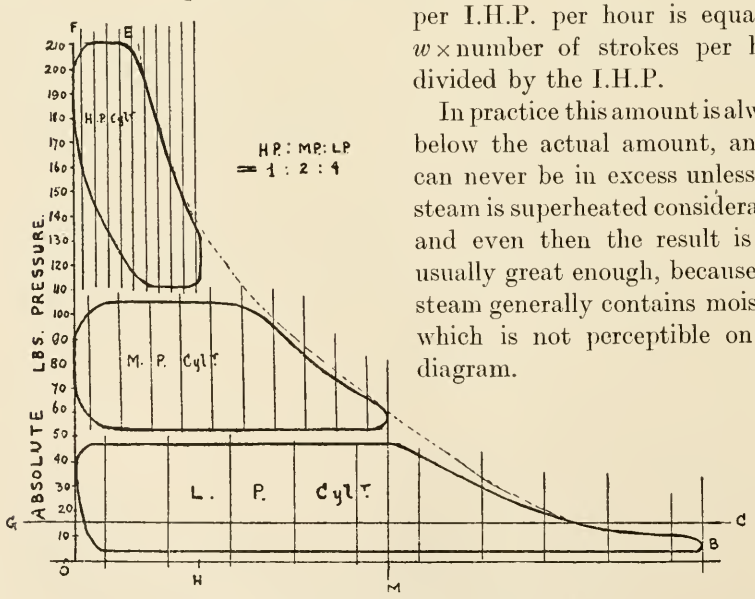


FIG. 123.—Combined Diagrams, neglecting Clearance.

Combined Diagrams of Stage Expansion Engines.—The diagrams taken from the various cylinders of a stage expansion engine

are generally on different scales, such as are suited to the pressures used in them. In combining them, therefore, the scale must be corrected for each when plotting them on the same curve or figure. The horizontal length of each figure must be proportioned to the cylinder volume, and the vertical height which represents the pressure must be proportioned to a certain scale. If there is more than one cylinder in one stage of the expansion, the volumes must be added together to represent the volume of the virtual single cylinder on the combined figure.

After the diagrams are obtained either by actual practice or by precise calculation, the procedure is graphical, and as follows:

1. *When Clearance is not taken into Account.*—A base line, *OB* (Fig. 123), is taken to represent the volume of the L.P. cylinder or cylinders; then mark off *OM* to represent to the same scale the volume of the M.P. cylinder, and *OH* to represent the volume of the H.P. cylinder. Draw a perpendicular, *OF*, representing the absolute pressure of the steam to scale, so many pounds to one inch. Then construct the figures from the diagrams, taking care that the length and height are on the new scale. The figures obtained will be something like those shown. This combined figure is very similar to what might be expected if the steam were cut off at *E*, and the expansion carried out in a single cylinder.

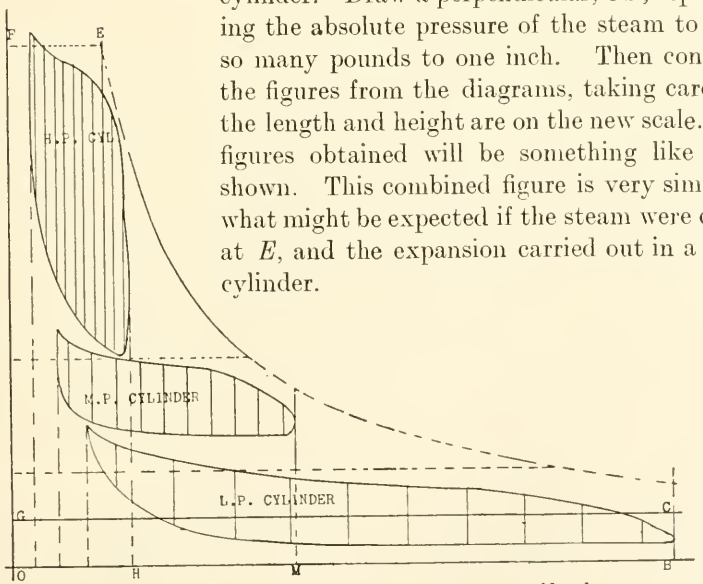


FIG. 124.—Combined Diagrams, Clearance considered.

2. *When Clearance is taken into Account.*—The actual amount of steam which is expanded in the H.P. cylinder, and so on through the various stages, is increased by the clearance spaces which must necessarily exist. Therefore, in setting out the various cylinder volumes, the actual volume of each cylinder *plus* the clearance is more

clearly correct. This is shown in Fig. 124. Each cylinder volume is credited with the addition due to its clearance volume.

3. *When Clearance and the Amount of Steam left in the Cylinder at the End of each Stroke are taken into Account.*—For a more precise method, the amount of steam remaining in the cylinder after compression begins must be deducted from the amount of steam in the cylinder at the instant of cut-off, and therefore a deduction should be made from the clearance steam for each cylinder, and it can be calculated by the method explained for the measurement of weight of steam used (Fig. 122). If the engine is " linked-up " considerably, the weight of steam added to fill the clearance volume is frequently reduced by one-half, and therefore in Fig. 124 the clearance spaces allowed must be reduced to the same extent. The conclusion is, therefore, that the combined diagram is somewhat between the extreme limits shown in Figs. 123 and 124, and the expansion curve, nearly hyperbolic, lies somewhere between the two curves shown.

Receiver Drop.—From all three methods it is evident that there is an apparent loss of work by using a three-stage expansion engine instead of expanding in a single cylinder, of the same volume as the L.P. cylinder. If the steam could be maintained in a state of dry saturation, and there were no cylinder condensation, the loss by using stage expansion would be very great; but it is found in practice, as explained elsewhere, that the great range in temperature obtained by expanding in a single stage produces excessive condensation loss, and thus the actual economy is greater with the stage expansion engine. In practice, if steam of similar initial pressure is expanded in a single stage instead of three stages, as shown in the figures, the forward pressure during nearly the whole of the stroke is lowered by warming the cylinder; and during the latter part of the stroke, corresponding to the L.P. cylinder in a three-stage engine, is but slightly greater than the back pressure, and in some cases actually below it.

Cippolini Continuous Double-diagram Indicator.—For close and accurate testing of reciprocating steam engines it is convenient to use a continuous automatic arrangement for taking the diagrams. In Fig. 125 a type of Dobbie-M'Innes external spring indicator is shown, adapted to the purpose of taking complete diagrams simultaneously and continuously from both ends of the engine cylinder. The string is attached to the engine piston rod in the usual way, and by an automatic arrangement records every revolution of the engine on a paper cylinder, *C*, which is also in connection with an electrical

timing arrangement, and thus the rate of revolution is also automatically recorded. The indicator can be set so as to record a reading

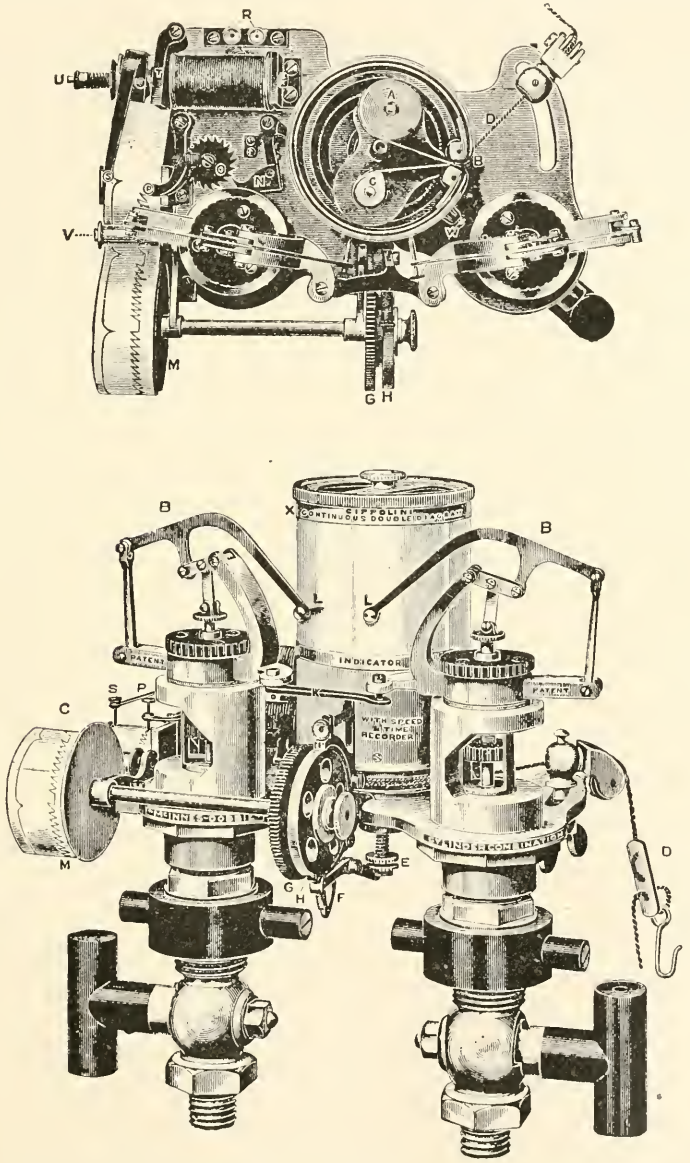


FIG. 125.—Cippolini Continuous Indicator (Dobbie M'Innes, Ltd.).

automatically every 25, 50, or 100 revolutions, or as required. The paper is contained within the barrel, and is automatically unwound from the drum at each period of registration.

The advantages of an automatic apparatus of this type, which avoids all personal error, are obvious. Each pair of diagrams can be taken either together or separately and entirely independent of the confusing lines of the usual continuous diagram. A full description is generally supplied with the apparatus, and it is not necessary to enter here on a more detailed description.

CHAPTER XIV

EFFICIENCY OF THE STEAM AND MECHANISM

Introductory.—In Chapter IV. a general idea of the laws of expansion under various assumed conditions has been given, and there now remain to be considered the variations produced in actual working, with their effects on the efficiency of the steam and their bearing on the efficiency of the engine.

The *mechanical efficiency* of the engine is almost entirely independent of the steam, and is governed particularly by the frictional resistances of the mechanism. These may or may not be affected by the increased temperatures and differences of temperature of the working parts and their proper lubrication, but under good conditions the difference in mechanical efficiency should be only very slight. For the present only the efficiency of the steam, which is the ratio of the useful work done by it in the engine to the total heat (in foot-lb. or b.t.u., according as the useful work is measured in the same units), conveyed to the engine from the boiler, is under discussion.

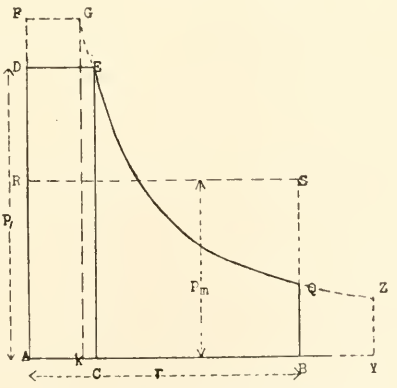


Fig. 126.—Influence of Boiler Pressure.

Influence of Boiler Pressure.—For convenience of demonstration of principle, the hyperbolic or other law of expansion may be assumed, as in Fig. 126. In this the

work done during admission = $1DEC$, and
work done during expansion = $CEQB$.

By increasing the ratio of expansion from $\frac{AB}{AC}$ to $\frac{AY}{AC}$, the quantity

of work done by the steam is increased, without admitting any more steam into the cylinder, and its efficiency is increased by using increased expansion.

By increasing the boiler, or admission, pressure from AD to AF, and cutting off admission at k, instead of at C, the work done during admission = $AFGk$, and the work done during expansion = $kGQB$. The area $AFGk = ADEC$, assuming hyperbolic expansion, and therefore the work done during admission is the same as before; but the ratio of expansion is increased, because steam is now cut off earlier in the stroke of the piston. More work is done by the steam, as represented by the area $DFGE$, and greater economy is obtained in its use and therefore greater efficiency, because only about the same quantity of heat is necessary to produce the higher initial pressure.

The increase in initial pressure does not increase the size of cylinder necessary for the same ratio of expansion, although more work is obtained. An increase in ratio of expansion, without increasing the initial pressure, increases the work done, but requires a larger cylinder in which to carry out the expansion; in other words, an increase in size of cylinder is necessary to obtain greater economy and efficiency, unless it is accompanied by an increase of pressure.

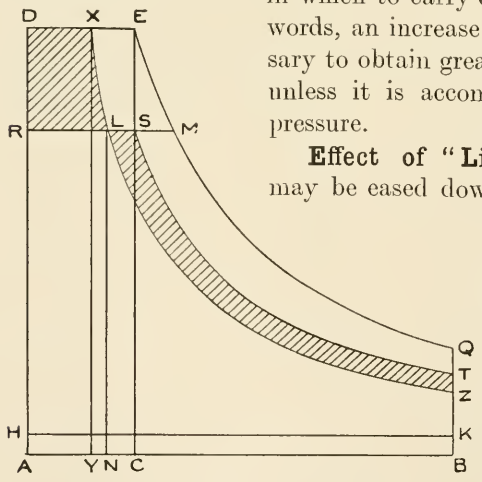


Fig. 127.—Throttling and Linking-up.

Effect of "Linking-up."—The engines may be eased down and the power decreased in two ways:

1. By *throttling* or partially closing the regulating valve and thus reducing the initial steam pressure AD to AR, as shown in Fig. 127. The cut-off and ratio of expansion are unaltered, but the volume of steam admitted per stroke is reduced from area $DECA$ to area $RSCA$, and its relative weight (at the same pressure AR) from RM to RS. Assuming KH to be the back pressure line, the work done per stroke is reduced from $DEQKH$ to $RSTKH$.

2. By *linking-up*, to obtain the same reduction of power as in (1), the original initial pressure AD may be retained and the ratio of expan-

sion increased. The weight and volume of steam per stroke would thus be reduced from *DECA* to *DXYA*. Assuming as in (1) *KH* to be the back pressure line, the work done per stroke is reduced from *DEQKH* to *DXZKH*. The work done, at reduced power, being equal in each of the two cases (1) and (2), then—

The area *RSTKH* = area *DXZKH* ; and therefore the area *DXLR* = area *LSTZ*, from which it is evident that *RL* must be less than *RS*, and consequently the weight and volume *RLNA* of steam used, when linked up, is less than the weight and volume *RSCA* of steam used when the regulating valve is throttled to obtain the same power. In practice the link is used as far as practicable, and the throttle as necessary after linking-up.

Cylinder Condensation.—At the beginning of the stroke of the piston after exhaust has taken place, steam is entering the cylinder. The cylinder and all parts of the metal which have been in contact with the exhaust have been cooled by its lower temperature, and consequently the entering steam does not act with its full energy, part of which is absorbed in reheating the cylinder metal. Condensation takes place, reducing the pressure, and the steam becomes moist ; the moisture is deposited on the walls of the cylinder, and the deposit increases the rate of conduction of heat from the steam to the walls, and thus tends to increase the loss.

There is also a slight condensation due to the conversion of some of the heat energy in the steam into work on the pistons ; but this liquefaction is partly counteracted by the re-evaporation during expansion, because the expanded steam requires less total heat to keep it in a state of saturation or dryness.

Complete and Incomplete Expansion.—For so long as the forward pressure (*OY* in Fig. 128) on one side of the piston is greater than the exhaust pressure *KL* on the opposite side, it is evident that more work can be obtained by further expansion. By increasing the expansion until the final pressure *KL* becomes almost equal to the exhaust pressure, the maximum work is obtained from the steam and the expansion is then complete. During the latter part of the stroke of the piston, however, very little work is added to that already done in the earlier part : because the difference of pressure on the two sides of the piston is so small, and may be insufficient to overcome the frictional resistance of the mechanism and, therefore, be powerless to produce motion. In reciprocating steam practice, expansion is seldom continued below 7 lb. absolute pressure, and is never entirely complete

for the above reasons, and also because the increased size of cylinder necessary for complete expansion is not actually economical in space and weight, as well as adding to the cylinder condensation losses. As can be seen from the indicator diagrams in the previous chapter, there is generally a rapid fall of pressure (to the exhaust pressure) towards the end of the stroke of the piston.

In the case of turbines, the expansion may be continued without inconvenience down to the absolute pressure of the condenser, and the increased work obtainable from the same steam is represented in the figure by the area $OBKX$, where OX represents the absolute pressure in the condenser.

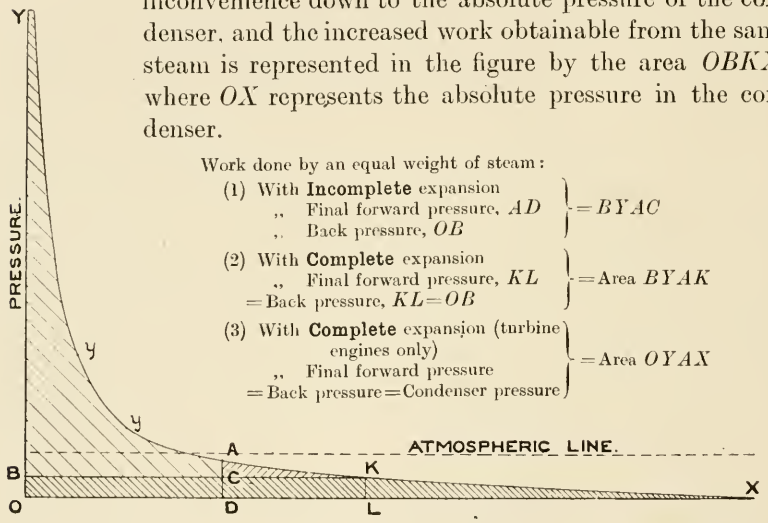


Fig. 128.—Complete and Incomplete Expansion. Work done by 1 lb. weight of Steam.

In the figure:

the area $BYAC$ = maximum work obtainable with incomplete expansion, final pressure AD, and back pressure $CD = KL$;

the area BYK = maximum work obtainable with complete expansion;

the area OYX = maximum work obtainable with turbines, back pressure line being OX.

Influence of High Vacua on Economy of Turbines. — The ratio of expansion in an ordinary reciprocating engine is the ratio of the volume occupied by the steam in the L.P. cylinder at the instant of release, to that occupied by the steam in the H.P. cylinder at the instant of cut-off. With a ratio of L.P. cylinder to H.P. cylinder of about 7 to 1, and a cut-off in the latter at about half-stroke, the

ratio of expansion is about 14, neglecting clearance; by linking up for low powers, the ratio is increased to about 20 to 24.

In the turbine the steam is expanded down to almost the exhaust pressure, as shown in Fig. 129,[1] and the ratio of expansion is practically governed by the ratio of the steam pressure on admission to the pressure of exhaust. Assuming that the admission pressure is 150 lb. absolute, and that the exhaust pressure is 2 lb. absolute, the ratio of expansion becomes about $\frac{150}{2}=75$. The final, or condenser, pressure is practically constant, and no increase of expansion is possible except by increasing the admission pressure.

At low powers the pressure of admission is reduced by throttling or wiredrawing and the ratio of expansion is *reduced* in the turbine; but in the reciprocating engine, as we have seen, the ratio of expansion may be actually *increased* by linking up, and consequently the economy of the turbine at low powers must be less than that of reciprocating engines, unless some special method is provided (see Chap. XXI.).

Steam Jacket.—To coun-

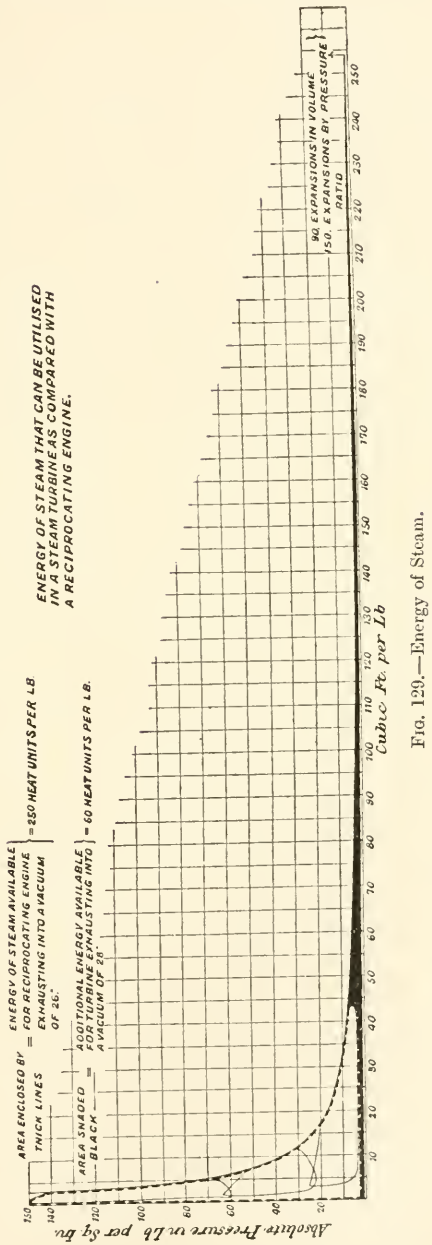

AREA ENCLOSED BY THICK LINES = ENERGY OF STEAM AVAILABLE FOR RECIPROCATING ENGINE EXHAUSTING INTO A VACUUM OF 26" = 250 HEAT UNITS PER LB.
AREA SHADED BLACK = ADDITIONAL ENERGY AVAILABLE FOR TURBINE EXHAUSTING INTO A VACUUM OF 28" = 60 HEAT UNITS PER LB.
ENERGY OF STEAM THAT CAN BE UTILISED IN A STEAM TURBINE AS COMPARED WITH A RECIPROCATING ENGINE.
90 EXPANSIONS IN VOLUME
150 EXPANSIONS BY PRESSURE
RATIO
Absolute Pressure in lb per Sq In
Cubic Ft. per Lb

Fig. 129.—Energy of Steam.

[1] From *Engineering*, October 5, 1906. Paper by Hon. C. A. Parsons and R. J. Walker.

teract the losses from cylinder condensation the steam jacket was introduced, and exists in nearly all large engines. The steam is admitted into a space between the liner and the cylinder, and any water produced by the transfer of heat to the steam in the cylinder is drained away as required.

Under certain conditions of working, the steam jacket produces great economy by keeping the steam in the cylinder dry, and thus performing efficient work through its medium. The steam jacket, however, must be given sufficient time to transfer its heat to the steam in the cylinder; with a high rate of revolution particularly, and with great piston speed indirectly, it ceases to be efficient, and may result in an actual loss of heat. The steam jacket, with short stroke engines, is seldom a source of economy when used for over one-fourth power, but no distinct line can be drawn for all engines; it is a case of each engine standing on its own merits.

The top and bottom ends of the cylinder are theoretically useful for steam jacketing, but in practice they are generally inoperative; this is due to the slag or core surface of the metal and to deposits of grease or oil, which prevent any rapid transference of heat. Thus jacketing the ends of the cylinders is of little use in producing economy, and at the same time heat is radiated from them into the engine room, producing unnecessarily high temperatures without any compensating useful effect.

The interior surface of the wall of the cylinder (*i.e.* the liner barrel) is kept in a clean and polished state by the reciprocation of the piston, and acts quickly as a conductor of heat, but it is obvious that the jacket heats the exhaust steam also. The quantity of heat transferred to the exhaust is generally very small, and not of very great consequence, due to the fact that there is little moisture present, and that the rate of conduction is very slow. Any heat imparted to the exhaust is a loss in a single-stage expansion engine, because the heat passes into the condenser without doing any work. It is probably for this reason that there is seldom any practical economy in jacketing the L.P. cylinder of a compound engine, although under the same conditions there may be a fair economy by jacketing the H.P. and M.P. cylinders.

Thermo-dynamically the L.P. jacket is the most efficient, and this would be the case practically if no re-evaporation occurred at the end of each stroke. Unfortunately, however, it is almost impossible to efficiently jacket this cylinder, so that steam is formed by

re-evaporation, and passes directly into the condenser instead of doing work in another cylinder.

If economy is to be gained by jacketing, the pressure in any one jacket must be at least as great as the initial pressure in that cylinder, or the jacket steam will extract heat instead of imparting it. This is arranged for in practice by fitting pressure gauges in connection with the jacket and the receiver, side by side for each cylinder, so that they can be readily seen and regulated.

Jackets are most useful for warming the engines in the first instance, and for keeping them warm when waiting orders; they appear to be particularly necessary for naval engines, except those of the smallest size, where small weight is of greater importance.

The table on the following page shows the result of jacketing the cylinders of H.M.S. *Argonaut* and *Diana*, and it should be noticed that it bears out the foregoing remarks, which were originally made before these trials had been carried out.

Superheated steam was extensively used to counteract cylinder condensation in marine engines when working pressures of from 5 to 35 lb. were common. With the advent of higher working pressures and, therefore, largely increased temperatures of steam, the difficulty of internal lubrication also greatly increased. With the higher flash-point oils now used for internal lubrication there is no difficulty in making lubrication efficient. Turbines require no internal lubrication, and the difficulty does not exist.

Dry steam does not quickly absorb heat which is supplied by the fuel, and consequently a much higher temperature must surround the tubes in which the steam is contained, or a larger heating surface be provided for the process of superheating than would otherwise be necessary, say, for water which absorbs heat very quickly. The early superheaters were fitted in the uptake, and when the temperature of the steam leaving the boiler seldom exceeded 280° F., a fairly large quantity of superheat could be added without increasing the funnel temperature (which under natural funnel draught only was generally above 600° F.), and without increasing the funnel loss. With the high pressures (235 lb.) and temperatures (402° F.) of steam now in common use, and the moderate funnel temperatures of 560° to 600° F. (frequently less with hot-air draught and with economisers), the difference which might be absorbed by the steam and used to superheat it, is less, but also useful.

A superheater to be efficient must be placed where the temperature

	I.H.P. Developed.	Fraction of Maximum Power.	Cylinder Jackets in Use.	Pressure of Steam.		Cut-off in H.P. Cylinder per cent.	Total Expansion. (Neglecting Clearance.)	Steam Consumption in lb. of Water used per I.H.P. per hour.		
				Boilers.	H.P. Receiver.			Main Engines.	Auxiliary.	Total.
H.M.S. *Argonaut.*	18,000	1	All	299	237	71	8·5	16·64	1·88	18·52
			L.P. only	290	242	71	8·5	16·15	1·88	18·03
			None	292	236	71	8·5	15·75	1·88	17·63
	13,500	3/4	All	280	244	43	12·5	15·85	1·59	17·44
			M.P. and L.P.	280	234	53	10·4	15·87	1·59	17·46
			None	251	230	53	10·4	15·44	1·59	17·03
	3,600	1/5	All	285	164	27	15·7	16·32	3·57	19·89
			M.P. and L.P.	294	166	27	15·7	16·34	3·57	19·91
			None	275	168	27	15·7	16·26	3·57	19·83
H.M.S. *Diana.*	8,000	5/6	All	151	145	65	6·4	17·75	1·93	19·68
			L.P. only	149	143	65	6·4	17·94	1·96	19·90
			None	151	144	65	6·4	17·57	1·93	19·50
	6,400	2/3	All	146	121	60	6·8	18·06	2·19	20·25
			M.P. and L.P.	136	119	60	6·8	17·91	2·65	20·56
			L.P.	136	117	60	6·8	17·84	2·66	20·50
			None	146	121	60	6·8	17·45	2·68	20·13
	4,800	1/2	All	148	128	42	8·5	16·54	1·72	18·26
			M.P. and L.P.	140	124	42	8·5	16·80	1·64	18·44
			None	139	127	42	8·5	16·45	1·75	18·20
	1,600	1/6	All	122	76	25	11·5	17·60	3·40	21·00
			H.P. and M.P.	122	78·5	25	11·5	17·70	3·30	21·00
			None	126	80·5	25	11·5	19·20	3·30	22·50

of the heated products of combustion is sufficiently high, and where the heat remaining, after effecting the process of superheating, does not increase the funnel loss. In installations, therefore, the position of the superheater may be either next the fire, where it is subjected to the greatest temperature, or in some intermediate position such as the combustion chamber of a cylindrical boiler or in a corresponding position in a water-tube boiler. In the mercantile marine, where a steady output of steam is maintained for long periods, superheating can be safely used without danger from a rise in temperature of the metal, generally mild steel, of the superheating tubes; in other words, the degree of superheat can be fairly well regulated when the output of steam is practically constant, and the economy of fuel and efficiency of the steam may be increased. The same conditions obtain in many land installations of power, and a corresponding economy of from 5 to 15 per cent results.

With turbine engines internal lubrication is unnecessary, and therefore no difficulties arise from it. The difficulty of efficient regulation of the amount of superheat still applies, and necessitates in reaction turbines an increased radial clearance between both the moving blades and the casing and between the fixed blades and the moving drum of the turbine. The larger clearances tend to increase the loss of steam (and energy), which escapes into the exhaust without doing useful work; but the rapid interchange of heat which takes place in the blade passages tends to counteract the clearance loss by absorbing moisture, and by maintaining a better quality of steam.

Fig. 130 shows the gain in economy due to superheat, and with modern turbines and superheaters this economy is now realised in practice, many of the early difficulties having been overcome by experience and experiment.

Stage Expansion.—Working expansively in a single cylinder is economical until a point is reached when the alternating temperatures in the cylinder become too great in range, so that a great part of the energy in the steam is lost through cylinder condensation and radiation from the internal and external surfaces of the cylinder. Generally a range of temperature of about 80° is the maximum that can be worked without serious loss; but at low pressures the losses are not so great, and ranges of from 100° to 120° are adopted with a fair economy. By dividing the expansion between two or more stages, in separate cylinders, the range of temperature in each is proportionately lowered, and this is done in practice by adopting stage-expansion, or compound, engines.

Economy, even with stage-expansion engines, can only be obtained when a proper range of temperature is adopted for each cylinder. Thus a two-stage engine, using 140 lb. pressure, shows little economy over one using 90 lb. pressure; while a three-stage engine shows a greater gain than theoretically due, when working at 140 lb. pressure against a two-stage engine working at 90 lb. The actual amount of

CURVE SHOWING GAIN IN ECONOMY DUE TO SUPERHEAT.
(Based on Tests of "Brown-Curtis" Experimental Turbine.)

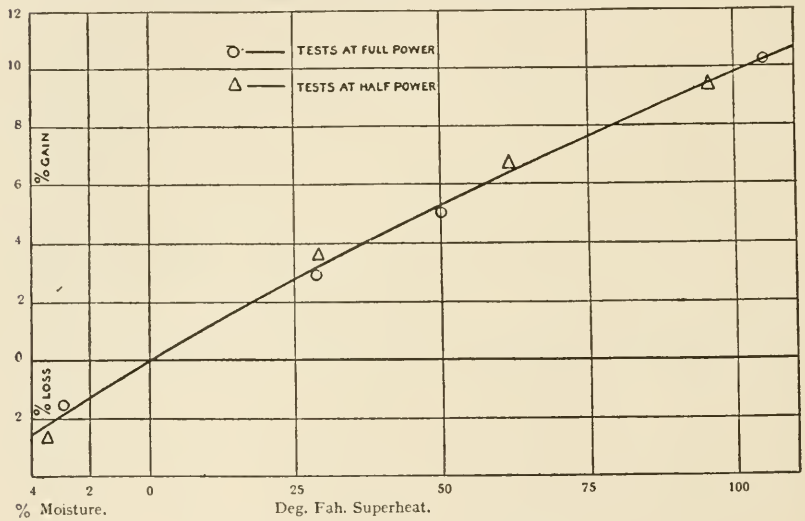


FIG. 130.—Economy due to Superheat.

fuel saved by using a three-stage engine working at 155 lb. pressure compared with a two-stage engine using 80 lb., is about 20 per cent.

For pressures over 180 lb. a four-stage engine is more economical in fuel than a lower-stage engine, and for mercantile purposes four stages are generally used for this economical reason.

Proportion of Cylinder Volumes.—The pistons have the same stroke in any one particular engine, and consequently the volumes of the cylinders are generally in exact proportion to their areas or the squares of the diameters of the cylinders; this last is the expression generally stated.

For naval engines the proportions adopted were as below:

Date, about.	Initial Pressure.	H.P. Cyl.	M.P. Cyl.	L.P.Cyl.
1890	155 lb. per square inch	1	2·25	5 to 6
1894	250 ,, ,,	1	2·66	7·01
1903	210 ,, ,,	1	2·5	6·28
1905	250 ,, ,,	1	2·8	7·9

The cut-off was at about 0·75 of the stroke, and this produced an apparent ratio of expansion of about 7 to 10, but is actually less, as the clearance spaces were large.

For the mercantile marine the following proportions are common :

Type of Engine. Reciprocating.	Initial Pressure.	H.P. Cyl.	M.P. Cylinder.		L.P. Cylinder.
Two-stage expansion	90 lb.	1	...		5·2
,, ,,	120 ,,	1	...		6·75
Three-stage expansion	120 ,,	1	2·1		5·4
,, ,,	140 ,,	1	2·2		6·2
,, ,,	160 ,,	1	2·4		7·0
,, ,,	180 ,,	1	2·5		7·8
			1st M.P.	2nd M.P.	
Four-stage expansion	180 ,,	1	1·8	3·6	7·8
,, ,,	215 ,,	1	2·05	4·22	8·62 (*Orontes*)
,, ,,	215 ,,	1	1·73	4·0	9·0 (*Agamemnon*)
,, ,,	267 ,,	1	2·0	4·0	12·2 (*Inchmarlo*)
,, ,,	210 ,,	1	2·0	3·9	7·9 (*Caronia*)

The cut-off adopted is generally at about 0·6 of the stroke, but there is some considerable variation in different engines.

Comparative Size of Single and Three-stage Expansion Engines.—For purposes of design, the whole of the power is considered to be obtainable from the L.P. cylinder, and this would be the case in practice if cylinder condensation losses did not exist. The H.P. and M.P. cylinders are merely adjuncts of the L.P. cylinder for the purpose of dividing the expansion into stages, and thus saving the losses from condensation.

As an example of comparative size, consider two three-cylinder engines of the same power, length of stroke, rate of revolution, and nominal ratio of expansion ; but one engine will exhaust from each cylinder directly into the condenser, while the other will take steam in series from cylinder to cylinder, and finally exhaust into the condenser. Then by a simple calculation it can be shown that if the diameter of each cylinder in the single-stage engine is 40 inches, the cylinders of

the three-stage expansion engine will be about 28, 44, and 72 inches in diameter—that is, the higher-stage expansion engine occupies more space than the single-stage expansion engine of the same power ; but as the coal consumption per I.H.P. is less in practice, the boiler power can be reduced, and generally a *total* saving of weight and space of machinery and boilers is obtained.

Efficiency of the Steam. — The table of maximum efficiencies of a perfect heat engine, given in Chapter V., shows that there is an increase—

1. By lowering the back or final pressure (and temperature).
2. By increasing the initial or boiler pressure (and temperature).
3. By superheating the steam before its admission to the engine (and thus increasing its temperature).

The increase of *actual* efficiency by the use of modern engines is much greater than would appear from the table, because the efficiency practically obtained more nearly approaches the maximum efficiency.

The application of the indicator by which the indicated horse-power is obtained gives the only real measure of the efficiency of the steam. For example, consider an engine working with an initial pressure of 210 lb. per square inch (temperature 389° F.), and with a feed-water temperature of 100° ; and that for each horse-power, as found by the application of the indicator, about 15 lb. weight of steam are required per *hour*. About 42·3 b.t. units are required for each horse-power, or 33,000 foot-lb. of work done per *minute*.

The useful work done by this 15 lb. weight of steam is (per hour)—

$$42{\cdot}3 \times 60 \text{ (minutes)} = 2538 \text{ b.t. units.}$$

The total heat supplied (net) per 60 minutes

$$= 15 \times \{966 - 0{\cdot}7(389 - 212) + (389 - 100)\}$$
$$= 15 \times 1131 = 16{,}975 \text{ b.t. units ;}$$

and the *actual efficiency* of the steam $= \dfrac{2538}{16{,}975} = 15$ per cent nearly.

A steam consumption of from 10 to 12 lb. per S.H.P. per hour is that usually obtained from warships ; but in the mercantile marine it is frequently less—from $9\frac{1}{2}$ to 11 lb. only is fairly common. Calculated as above, the steam efficiency of a turbine using 10 lb. of superheated steam per I.H.P.-hour

$$= \frac{42{\cdot}3 \times 60}{10 \times 1131} = 22{\cdot}5 \text{ per cent nearly.}$$

This may be considered as about the *maximum actual* efficiency of the steam at present attainable, and is about 46 per cent of the maximum efficiency of a steam engine considered as a perfect heat engine.

Summary of Heat Losses.—The following estimate of the actual losses of heat is taken from experimental results obtained from a good and fairly efficient steam plant. The figures given are based on the assumption of a supply of 100,000 b.t. units from the fuel burnt :

Description of Loss.	Unpreventable Loss per cent.	Actual Engine per cent.	Approximate per cent.
Boiler radiation and leakage	...	5·450	$5\frac{1}{2}$
Radiation of flue gases on way to economiser	...	0·545	$\frac{1}{2}$
Radiation from economiser	...	2·725	$2\frac{3}{4}$
Up the funnel	8·540	10·980	11
Radiation from feed pipe to boiler	...	0·135	$\frac{1}{4}$
Total boiler losses	8·540	19·835	20
Steam pipes, radiation from	...	1·685	$1\frac{1}{2}$
Radiation from jacket drain pipes	...	0·083	...
Radiation from engine	...	2·452	$2\frac{1}{2}$
Engine friction	...	1·000	1
Total engine losses	...	3·535	$3\frac{1}{2}$
Radiation from eduction pipe	...	0·925	1
Radiation from feed arrangements	...	1·060	1
Heating the circulating water	65·220	59·150	59
Total condenser losses.	65·220	61·135	61
Total losses	73·760	86·190	86
Leaving useful work	26·24	13·810	14

When temperature of steam at admission is 359° Fahr.
,, ,, ,, exhaust is 100° ,,
,, ,, feed water to economiser is 81° ,,
,, ,, ,, to boiler is 278° ,,
,, ,, jacket water entering boiler is 328° ,,
,, ,, ,, leaving engine is 335° ,,

Various Efficiencies.—The figures in the above summary, although true in this particular experiment, are not conclusive evidence of the actual losses in any other instance. They can, however, serve the purpose of showing how the various efficiencies of an actual engine are calculated.

(1) The *total or overall* efficiency of the engine takes into account

the engine friction, and therefore when 13,810 out of 100,000 b.t. units are converted into useful work—

$$\text{Efficiency} = \frac{13{,}810}{100{,}000} = 13{\cdot}81 \text{ per cent} \left(= \frac{\text{Shaft energy}}{\text{Fuel energy expended}} \right).$$

(2) 19,835 units are lost in various ways from the boiler, leaving 80,165 actually in the steam on its exit from the boiler. 1685 units are lost from the steam pipe before the steam reaches the engine, and thus leave a *net* supply of heat in the steam to the engine of 78,480 units. Including the steam pipe losses with those in the boiler—

$$\text{Efficiency of the } \textit{Boiler} = \frac{78{,}480}{100{,}000} = 78{\cdot}48 \text{ per cent.}$$

(3) From the 78,480 units supplied to the engine in the steam, radiation from the jacket drain pipes and from the engine take away 2535 units and the condenser losses (or exhaust waste) absorb 61,135 units, thus making a total loss of heat in the steam of 63,670 units—

$$\text{Efficiency of the } \textit{Steam} = \frac{78{,}480 - 63{,}670}{78{,}480} = \frac{14{\cdot}81}{78{\cdot}48}$$
$$= 19 \text{ per cent nearly.}$$

(4) Of the 14,810 units converted into useful work on the pistons, 1000 are absorbed in engine friction—

$$\textit{Mechanical} \text{ efficiency of the } \textit{Engine} = \frac{14{\cdot}81 - 1}{14{\cdot}81} = 93{\cdot}24 \text{ per cent.}$$

(5) The total or overall efficiency of the engine is the product of the *three* efficiencies of the boiler, the steam, and the mechanism respectively—

$$= \frac{78{\cdot}48}{100} \times \frac{14{\cdot}81}{78{\cdot}48} \times \frac{13{\cdot}81}{14{\cdot}81} = 13{\cdot}81 \text{ per cent, which agrees with (1).}$$

(6) The *maximum efficiency, considered as a perfect heat engine*, does not take into account the boiler and engine friction losses, but only those of the steam as a working substance between its temperature limits. This efficiency is thus only comparable with the efficiency of the steam, as given in (3)—

$$\text{Maximum efficiency} = \frac{T - t}{T + 461} = \frac{359 - 100}{359 + 461} = \frac{259}{820}$$
$$= 31{\cdot}6 \text{ per cent nearly.}$$

(7) Compared with the maximum efficiency, the steam efficiency of the above engine—

$$= \frac{19}{31{\cdot}6} = 60 \text{ per cent about of the maximum possible efficiency.}$$

(8) The *thermal efficiency* of a steam plant—

$$= \frac{\text{heat converted into indicated work}}{\text{heat supplied to boiler}},$$

which in the above example $= \frac{14{,}810}{100{,}000} = 14{\cdot}81$ per cent.

The thermal efficiency is the product of the boiler and steam efficiencies, and in practice is generally calculated from the coal or fuel used per I.H.P. per hour, when the calorific value of the fuel is known.

EXAMPLE.—An engine uses $1\frac{1}{2}$ lb. of coal per I.H.P.-hour, and the calorific value of the coal is 15,000 b.t. units per lb.—

$$1 \text{ I.H.P.-hour} = \frac{33{,}000}{780} \times 60 \text{ (minutes)} = 2538 \text{ b.t. units per hour.}$$

$$\text{Thermal efficiency} = \frac{2538}{1\frac{1}{2} \times 15{,}000} = 11{\cdot}28 \text{ per cent.}$$

Comparative Thermal Efficiencies of Heat Engines.—Taking the formula for the maximum efficiency of a perfect heat engine as a basis of comparison :

A *steam* engine receiving steam at a temperature of 486° F., and exhausting it at 113° F. into a condenser—

$$\text{Efficiency} = \frac{486 - 113}{486 + 461} = \frac{373}{947} = 39{\cdot}4 \text{ per cent nearly.}$$

A *gas* or *oil* engine working between the limits of 2800° F. and 60° F. (exhausting into the atmosphere)—

$$\text{Efficiency} = \frac{2800 - 60}{2800 + 461} = \frac{2740}{3261} = 84 \text{ per cent.}$$

Of which in each case about one-half may be realised in the best practice.

Efficiency of the Mechanism.—The invention of the flash-light and other torsion meters has enabled engineers to study mechanism efficiency from an entirely new and reliable standpoint. Previously the method of estimating shaft H.P. was almost entirely empirical, and

based on generalities and theoretical estimation rather than on individual basis. One of the most instructive matters in connection with mechanical efficiency is described below, and reference should be made to shaft H.P. in Chapter V., and to certain theoretical considerations on the pressure on the guide (Chapter XVI.).

It is now proposed to compare the crank effort produced by the actual effective pressure in the cylinder with that observed by a torsion meter. In Fig. 131 the crank circle is divided into 24 equal parts, like a 24-hour (continental) clock, and the actual crank effort, or turning moment, at each point is then plotted on a curve, shown in full lines in Fig. 131A, for the L.P. cylinder. Suppose the crank is in any position, AC, and the connecting rod is then represented by BC, produced. Then, the turning moment, or crank effort, at C = force exerted along $BC \times AM$. The effective pressure in the cylinder can be obtained from the indicator diagram, as shown, in which the various positions of measurement (corrected for obliquity) are made to correspond with the position of the piston in the cylinder.

Let the effective pressure at position 2 be represented by P in lb., and of this force a certain proportion is exerted on the guide so that, graphically represented, the force acting along the line

$$BC = \frac{BM}{AB} \times P \, . = \frac{BC}{BD} \times P \, .$$ (where CD is perpendicular to BC).

The crank effort at 2, therefore,

$$= P \times \frac{BC}{BD} \times AM.$$

By adopting a connecting-rod length of 4 cranks, the usual practice, a table of values at each of the 24 points for BD and AM can easily be formed, and the curve shown in Fig. 131A can be plotted. The one shown in the figure has been obtained by graphic measurement and slide-rule calculation, and may not be exactly accurate, but it shows clearly the salient points. Two dotted curves are also shown which represent the crank efforts of the H.P. and M.P. respectively ; the three cranks being disposed at equal angles of 120 degrees, the three curves are plotted correspondingly. The combined efforts on the three cranks are next added together and a new curve set up from the same base line ; this curve is the upper one shown in full lines. The average effort so found is shown by the straight full line parallel to the base. From these curves it will be seen that the combined effort of a three-

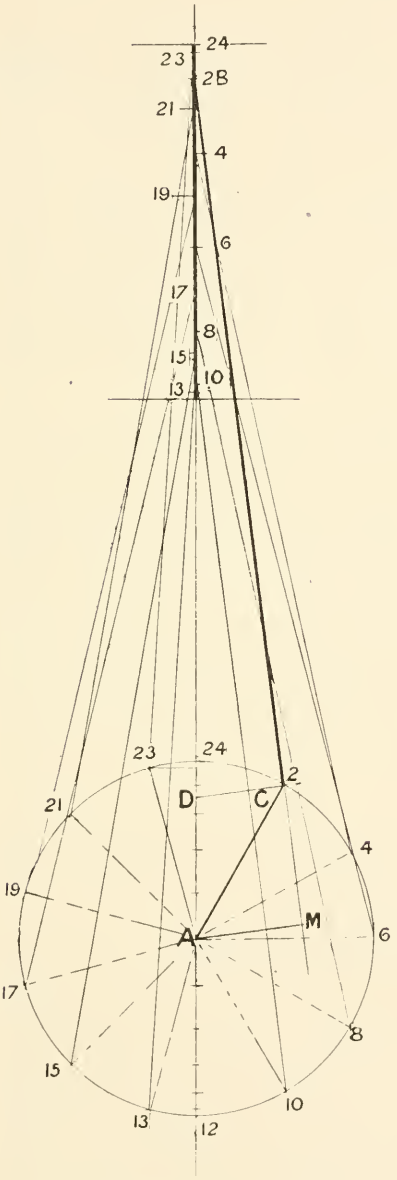


FIG. 131.—Calculation of Crank Effort from Indicator Diagram.

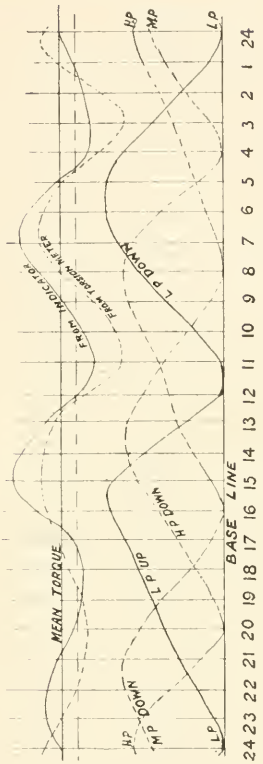
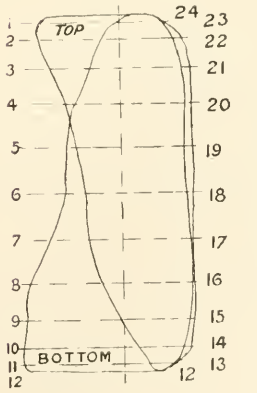


FIG. 131A.—Curves of Crank Effort, or Torque. (Not including momentum.)



Indicator Diagram.

crank engine is by no means uniform, and this is true of any expansive working engine with any number of cranks.

Below the upper curve another is shown in dotted lines, corresponding to the actual torque, as determined by the flash-light torsion meter, for the same engine working under the same conditions of power and revolution. From this curve it will be seen that still greater variation (maximum and minimum) is obtained than appears in the indicator measurement curve. The average torque is shown by the straight dotted line. The difference in the heights of these two straight lines above the base line represents the proportion (about $8\frac{1}{4}$ per cent) of the I.H.P. expended in overcoming the frictional resistances of the engine mechanism, which in this particular case includes the thrust friction, because the shaft H.P. torque was measured abaft the thrust block.

For a turbine engine the torque is practically uniform, and any variation points to defective balancing of the rotating parts or to unequal work done by the several blades of the propeller. Incidentally, the various curves show that equal powers transmitted through identical shafting produce considerably less maximum stress when the shaft is rotated by a turbine than when rotated by a reciprocating engine, however well it may be proportioned to obtain as uniform a turning moment as possible. With a reciprocating engine the fluctuation of torque produces corresponding variations in the twist of the shaft during every revolution and thus tends to reduce the life of the shaft by continual change in stress and shape, which may eventually end by fracture.

In Chapter XXVII. a new and accurate method of measuring the propeller thrust is illustrated and described ; this gives a further opportunity for testing efficiency of propulsion and propellers, generally, and including actual hull resistance.

Horse-power developed per Ton of Fuel.—The fuel consumption per I.H.P. or S.H.P. is generally expressed in lb. per hour, which as a small decimal fraction does not convey to the mind the real contrast of power developed from the fuel. This may be brought into greater contrast by expressing the same thing differently as shown in the annexed table. For example, a common pre-war figure was 1·55 lb. per S.H.P.-hour ; it is now, say, about 1·15 lb. From the table, 1445 S.H.P. per ton was obtained, which has now been increased to about 1948 S.H.P.-hour.

Table

Fuel burnt per I.H.P. per Hour.	I.H.Power developed per 1 Ton of Fuel burnt per Hour.									
lb.	0·00	0·01	0·02	0·03	0·04	0·05	0·06	0·07	0·08	0·09
0·8	2800	2766	2732	2696	2664	2632	2602	2574	2544	2512
0·9	2480	2456	2432	2410	2386	2358	2334	2308	2284	2262
1·0	2240	2217	2194	2172	2150	2133	2113	2093	2094	2056
1·1	2037	2019	2000	1983	1966	1948	1932	1916	1900	1884
1·2	1867	1852	1837	1822	1807	1792	1778	1764	1750	1736
1·3	1723	1720	1707	1695	1683	1660	1648	1636	1624	1612
1·4	1600	1589	1578	1567	1556	1545	1534	1523	1512	1502
1·5	1493	1483	1473	1464	1454	1445	1435	1426	1417	1408
1·6	1400	1392	1383	1374	1366	1357	1348	1340	1332	1324
1·7	1316	1308	1301	1294	1287	1280	1272	1264	1256	1248
1·8	1240	1234	1228	1222	1216	1211	1205	1199	1193	1186
1·9	1179	1173	1167	1161	1154	1148	1142	1136	1131	1125
2·0	1120	1114	1108	1105	1099	1093	1087	1081	1076	1071
2·1	1066	1061	1058	1050	1046	1041	1036	1031	1026	1020
2·2	1016	1014	1009	1004	1000	995	990	986	982	978
2·3	974	969	965	961	957	953	949	945	941	937
2·4	933	929	925	921	917	914	910	906	902	899
2·5	896	892	888	885	882	878	875	871	868	865
2·6	861	857	854	851	848	845	841	838	835	832
2·7	829	826	923	820	817	814	811	808	805	802
2·8	800	797	794	791	788	786	783	780	777	774
2·9	772	769	766	764	762	759	756	753	751	748
3·0	746	743	741	738	736	734	732	730	728	726

CHAPTER XV

ELEMENTARY THERMO-DYNAMICS

In the early chapters some idea is given of the nature and properties of heat, and its general relation to the working of the steam engine. In the present chapter it is proposed to carry this idea a step further and to develop certain details relating to the internal economy of heat engines, including the compression of air, compressed-air engines, and steam engines in general. The symbols which are used have been purposely chosen as being those usually found in works on Heat and Heat Engines, and, for convenience of reference, are tabulated below :

r	Ratio of expansion.
e	Napierian base of logarithms.
T_0, t_0	Absolute temperatures.
P, p	Absolute pressures in lb. per 1 square *foot*.
Pm	Mean absolute pressure ,, ,,
V, v	Volumes, in cubic feet, of 1 lb. of the substance.
Kp	Specific heat at constant pressure.
Kv	Specific heat at constant volume.
y	Ratio of the above specific heats $= \frac{K_p}{K_v}$.
c	Constant for a particular substance under discussion.
S	Sensible heat per 1 lb. of steam.
L	Latent heat of evaporation per 1 lb. of steam.
R	Internal latent heat of evaporation of steam.
x	Dryness fraction of steam.
H, h	Total heat of formation of 1 lb. of steam.

Summary of previous Formulae, etc., relating to heat and heat engines :

Boyle's Law : $V \propto \frac{1}{P}$, when T_0 is constant,

Charles' Law : $P \cdot V = c \cdot T_0$.

External work done $= p \cdot (V - v)$ (in ft.-lb. for each 1 lb. of the working substance).

Sensible heat : $S = T - t$. per 1 lb. of the working substance.

Maximum efficiency of an engine, using a perfect gas as the working substance only—

$$= \frac{T_0 - t_0}{T_0}.$$

Indicated horse-power developed in a single cylinder $= \frac{2 \cdot PLAN}{33{,}000}$

(with 2N working strokes per minute).

First law of thermo-dynamics :

Mechanical equivalent of heat, $J = 780$ ft.-lb. $= 1$ British thermal unit.

Second law of thermo-dynamics :

$$\text{Efficiency of conversion} = \frac{\text{heat converted into work}}{\text{heat taken in by the engine}}.$$

Latent heat of evaporation : $L = 966 - 0{\cdot}7(T - 212)$, per 1 lb. of saturated steam.

Total heat of evaporation : $H = 966 - 0{\cdot}7(T - 212) + (T - t)$

$= L + S$, per 1 lb. steam.

Total heat of formation of 1 lb. of steam, when dryness fraction $= x$:

$= x \cdot L + S$

$= x \cdot \{966 - 0{\cdot}7(T - 212)\} + (T - t)$.

Equivalent evaporation, *from* and *at* 212° F. $= \frac{n \cdot (x \cdot L + S)}{966}$, where

$n =$ lb. weight of steam at a given temperature, T, produced from a known temperature (t) of feed water.

Expansion of steam :

Hyperbolic curve, $P \cdot V = $ constant (T changes in value).

Saturation curve, $P \cdot V^{1{\cdot}0645} = $ constant.

Adiabatic curve, $P \cdot V^{1{\cdot}135} = $ constant, for steam initially dry and saturated.

$P \cdot V^{1{\cdot}035 + \frac{1}{10}} = $ constant, for wet steam of dryness fraction x.

Work done during Hyperbolic Expansion.—In Fig. 126 the area of *ADEQB* can be represented by some rectangular area, *ARSB*, when the height *AR* or *BS* is equivalent to the mean height of *DEQ* or the mean pressure ; call this p_m.

Let AD be called p_1 = initial pressure (absolute).
AB be called r = ratio of expansion

$$= \frac{\text{AB}}{\text{AC}}, \text{ when C is the point of cut-off.}$$

Then the work done during admission = area ADEC
$= p_1 \times \text{AC},$
and the work done during expansion = area CEQB

$$= \int_1^r p \times dv.$$

Now, $p \,.\, v = \text{constant} = p_1 \times 1$, for unit volume.

And, therefore, total work done $= p_1 \times 1 + \int_1^r \frac{dv}{v} \cdot p_1$

$$= p_1 + p_1 \log_e r$$
$$= p_1(1 + \log_e r).$$

$$\text{Now, } p_m = \frac{\text{area ARSB}}{\text{AB}} = \frac{\text{area ADEQB}}{r}$$

$$= \frac{\text{work done}}{r}.$$

$$\text{Therefore } p_m = p_1\left(\frac{1 + \log_e r}{r}\right).$$

Fig. 126 also illustrates how economy can be obtained —

(*a*) With increased expansion. If the rate of expansion is increased, represented by *AY* instead of *AB*, then the area representing the work done during expansion is *CEZY* instead of *CEQB*, an increase without the expenditure of more steam.

(*b*) With increased pressure. If the initial pressure is increased, represented by *AF* instead of *AD*, and the area *AFGk* is made equal to *ADEC* (the work done during admission), there is the same work done during admission ; but there is an increased amount, *kGQB*, done during expansion instead of *CEQB*, an increase with a very slightly greater expenditure of heat to obtain the higher initial pressure.

The increase in initial pressure does not increase the size of cylinder necessary for the same ratio of expansion, although more work can be done by the increased pressure.

The increase of the ratio of expansion without increasing the initial pressure, although it increases the work done, involves the use of a larger cylinder to carry out the expansion ; in other words, an increase

in size of engine is necessary to obtain greater economy unless accompanied by an increase of pressure.

Specific Heat.—The specific heats of the solid and liquid substances commonly used in marine engineering practice are given in Chapter III. In the following table the specific heats of some of the principal gases are now shown :

Name of Gas.	Specific Heat at Constant Pressure, K_p.	Specific Heat at Constant Volume, K_v.	Ratio of Specific Heats, $\gamma = \frac{K_p}{K_v}$.	
Air, dry	0·238	0·169	1·4	
Carbon dioxide, CO_2	0·217	0·172	1·26	
Carbon monoxide, CO	0·245	0·173	1·41	
Hydrogen, H_2	3·409	2·412	1·41	
Marsh gas, CH_4	0·593	0·47	1·26	
Nitrogen, N_2	0·244	0·173	1·41	
Olefiant gas, C_2H_4	0·404	0·359	1·12	
Oxygen, O_2	0·218	0·155	1·4	
Products of combustion—				
(Funnel gases)	0·24	0·17	1·4	Considerable variation in these values.
(Coal gas engine).	0·258	0·189	1·37	
Steam, superheated and above 212° F.	0·508	0·37	...	

A gas has two specific heats : the specific heat at constant *pressure* and the specific heat at constant *volume*. The general idea may best be illustrated by examples in common practice :

1. *When under Constant Pressure.*—In boiler furnace combustion the pressure of the air and of the products of combustion is very nearly atmospheric, and may be taken as constant : the specific heat at constant pressure of the products may be taken as about 0·25.

Suppose that 1 lb. of fuel gives out 15,000 b.t. units when burnt, and requires 19 lb. of air for its combustion in a furnace ; then the products of combustion weigh $1 + 19 = 20$ lb., and the heat given out by combustion is distributed among these 20 lb.

The rise of temperature of each 1 lb. of products for each 1 b.t.u.

$$= \frac{1}{0{\cdot}25} = 4^\circ \text{ F.}$$

The rise of temperature of 20 lb. of products for 15,000 b.t.u.

$$= \frac{15{,}000}{20 \times 0{\cdot}25} = 3000^\circ \text{ F. about.}$$

2. *When under Constant Volume.*—In a closed cylinder the volume

cannot increase and is, therefore, constant ; the specific heat at constant volume may be taken as about 0·19, when liquid fuel is used as the combustible.

Suppose that 1 lb. of liquid fuel gives out 19,000 b.t. units and requires 14 lb. of air for its combustion ; then the products of com-

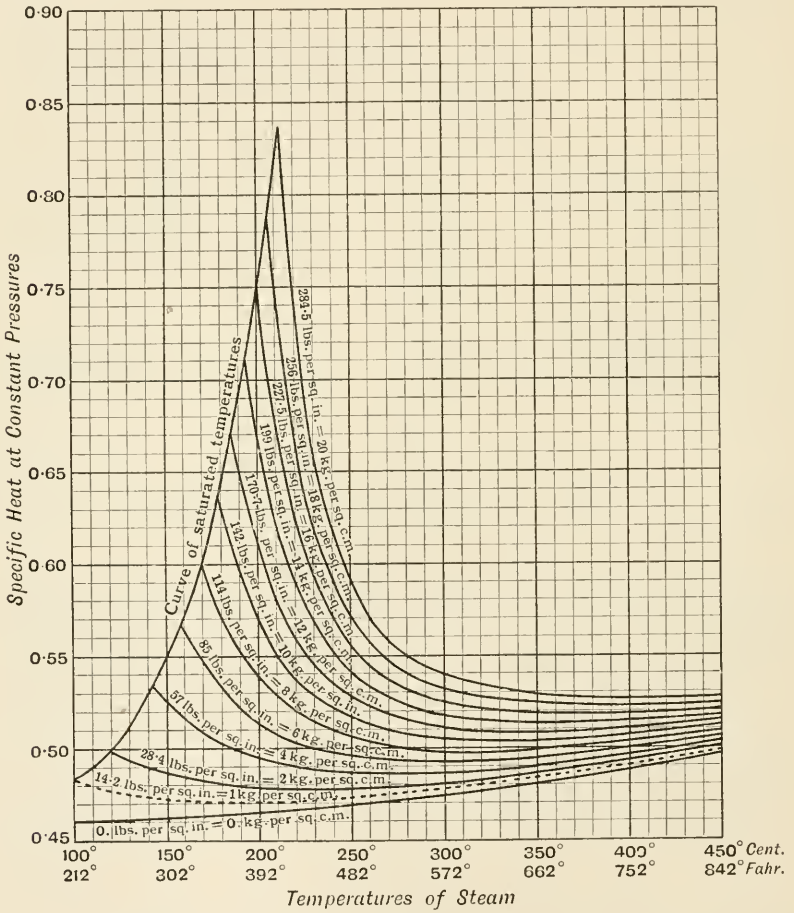

Specific Heat at Constant Pressures
0·90
0·85
0·80
0·75
0·70
0·65
0·60
0·55
0·50
0·45
Curve of saturated temperatures
293·5 lbs. per sq. in. = 20 kg. per sq. c.m.
256 lbs. per sq. in. = 18 kg. per sq. c.m.
227·5 lbs. per sq. in. = 16 kg. per sq. c.m.
199 lbs. per sq. in. = 14 kg. per sq. c.m.
170·4 lbs. per sq. in. = 12 kg. per sq. c.m.
142 lbs. per sq. in. = 10 kg. per sq. c.m.
114 lbs. per sq. in. = 8 kg. per sq. c.m.
85 lbs. per sq. in. = 6 kg. per sq. c.m.
57 lbs. per sq. in. = 4 kg. per sq. c.m.
28·4 lbs. per sq. in. = 2 kg. per sq. c.m.
14·2 lbs. per sq. in. = 1 kg. per sq. c.m.
0 lbs. per sq. in. = 0 kg. per sq. c.m.
100° 212°
150° 302°
200° 392°
250° 482°
300° 572°
350° 662°
400° 752°
450° Cent. 842° Fahr.
Temperatures of Steam

Fig. 132A.—Specific Heat of Steam at Constant Pressure (Kp).

Note.—The curves in Fig. 132B show the *mean* value of Kp between the saturated steam temperature and the final superheated temperature, both values being based on Fig. 132A. Thus, from Fig. 132B values, the total heat required for any degree of superheat can be easily calculated. Kp is a maximum at saturation point at all pressures above atmosphere, it then decreases until it reaches a minimum at a superheated temperature, and above this minimum Kp increases at still higher superheated temperatures.

bustion weigh 1 + 14 = 15 lb., and the heat given out by combustion is distributed between these 15 lb. of products.

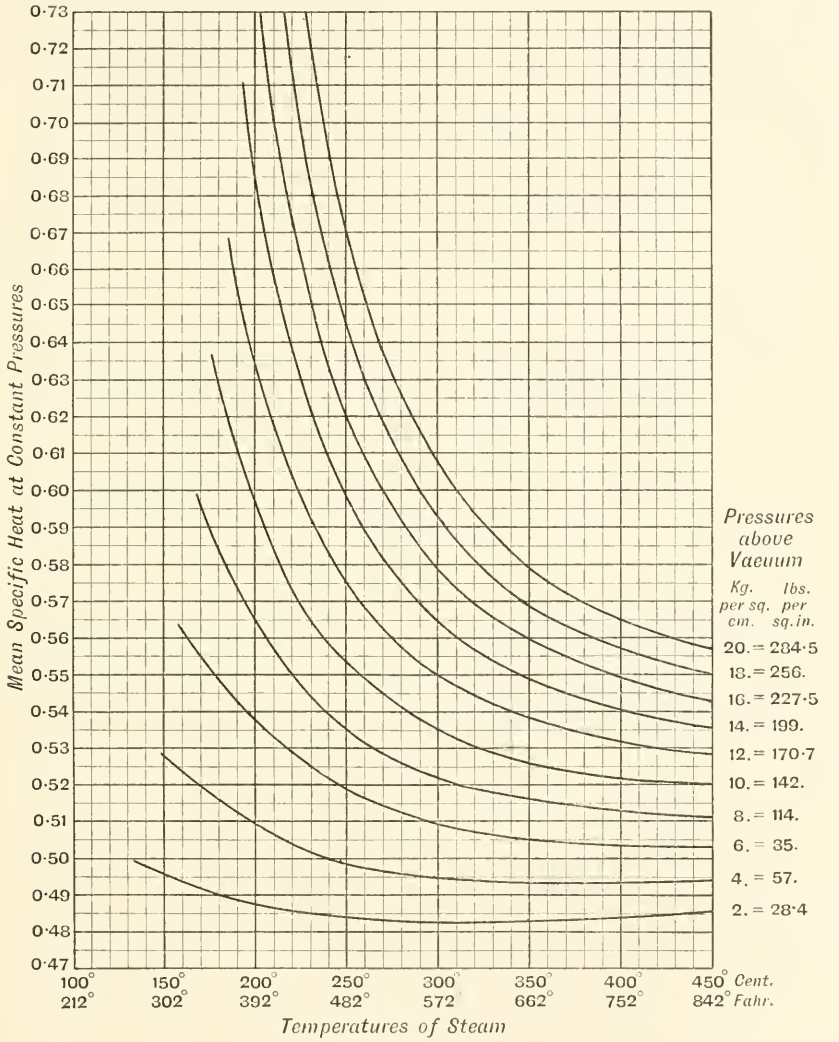


FIG. 132B.—Mean Specific Heat of Steam at Constant Pressure (Mean of Kp).

Note.—The curves in Fig. 132B show the *mean* value of Kp between the saturated steam temperature and the final superheated temperature, both values being based on Fig. 132A. Thus, from Fig. 132B values, the total heat required for any degree of superheat can be easily calculated. Kp is a maximum at saturation point at all pressures above atmosphere, it then decreases until it reaches a minimum at a superheated temperature, and above this minimum Kp increases at still higher superheated temperature.

The rise in temperature of each 1 lb. of products for each 1 b.t.u.

$$= \frac{1}{0{\cdot}19} = 5{\cdot}26^\circ \text{ F.}$$

The rise in temperature of 15 lb. of products for 19,000 b.t.u.

$$= \frac{19{,}000}{15 \times 0{\cdot}19} = 6667^\circ \text{ F.}$$

Definitions.—Specific heat at constant pressure (Kp) is the quantity of heat taken in by 1 lb. of a substance when its temperature rises 1° F., while the pressure remains unchanged and the volume is allowed to change.

Specific heat at constant volume (Kv) is the quantity of heat taken in by 1 lb. of a substance when its temperature rises 1° F., while the volume remains unchanged and the pressure is allowed to change.

Laws of the Permanent Gases. Third Law (Regnault).—The specific heat at constant pressure (Kp) is constant for any gas.

The approximate truth of this law has been demonstrated by experiment, and simply means that the heat taken in by 1 lb. of the substance is the same whether the rise of temperature be from, say, 32 to 33°, or from, say, 400 to 401° F., so long as the difference is exactly 1° F.

At constant pressure, the external work done due to a change of volume from v to $\mathrm{V} = p \cdot \left(\frac{\mathrm{V} - v}{\mathrm{J}}\right)$; and if T and t be the corresponding temperatures, the heat taken in by 1 lb. of the substance

$$= \mathrm{K}p \cdot (\mathrm{T} - t).$$

The increase in internal energy $= \mathrm{K}p \cdot (\mathrm{T} - t) - p \cdot \left(\frac{\mathrm{V} - v}{\mathrm{J}}\right)$.

From Charles' Law, $p \cdot \mathrm{V} = c \cdot \mathrm{T}_0$, and $p \cdot v = c \cdot t_0$;

and thus $$p\left(\frac{\mathrm{V} - v}{\mathrm{J}}\right) = c(\mathrm{T}_0 - t_0).$$

Therefore, increase in internal energy per 1 lb. of the substance

$$= \mathrm{K}p \cdot (\mathrm{T} - t) - c \cdot (\mathrm{T} - t)$$
$$= (\mathrm{K}p - c)\ (\mathrm{T} - t).$$

That is, the change in internal energy is exactly proportional to the change of temperature, when the pressure is constant.

At constant volume (when no external work is done) any heat taken

in by the substance produces a change in temperature and pressure, and $P \propto T_0$. The heat taken in by 1 lb. of the substance $= Kv \,.\, (T - t)$, and this must evidently = the increase in internal energy of each 1 lb. of the substance. That is, the change in internal energy is exactly proportional to the change of temperature, when the volume is constant.

Therefore, when the temperature of a substance is raised from t to T, the increase in internal energy at constant pressure = the increase in internal energy at constant volume ; and

$$(Kp \,.\, - c)\ (T - t) = Kv \,.\, (T - t)$$

From which, $$Kv = Kp - c,$$

and $$Kp - Kv = c.$$

Fourth Law (Joule).—When a gas expands without doing external work, and without either taking in or giving out heat (and, therefore, without changing its intrinsic energy), its temperature does not change.

This is a continuation of the application of Boyle's Law—

$$p \,.\, v = P \,.\, V = \text{constant, when T is constant.}$$

From the third law, it may now be written :

$$\frac{p \,.\, v}{t_0} = c = \frac{P \,.\, V}{t_0} = c = Kp - Kv.$$

The fourth law, like the others, is not exactly true except of a perfect gas. The difference is extremely small, but is distinctly apparent in the adiabatic expansion of steam particularly, and of air ; in each case there is a less pressure at the end of the expansion than would be shown by exact calculation based on the absolute truth of the law as enunciated.

Values of Kp, Kv, and y. For *dry air* :

As measured by Regnault, the value of

$$Kp = 0{\cdot}2375 \text{ b.t. units.}$$

Assuming that 1 b.t.u. $= 780$ ft.-lb. ; and $c = 53{\cdot}18$ (p. 26).

Then $$Kp = 184{\cdot}7 \text{ ft.-lb.}$$

From which, $$Kv = Kp - c = 184{\cdot}7 - 53{\cdot}18$$
$$= 131{\cdot}5 \text{ ft.-lb} = 0{\cdot}1696 \text{ b.t. units.}$$

And, $$y = \frac{Kp}{Kv} = \frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}$$
$$= \frac{0{\cdot}2375}{0{\cdot}1696}$$
$$= 1{\cdot}4 \text{ about.}$$

For *superheated steam*, whose density under similar conditions of pressure and temperature is only 0·622 times that of dry air, the different value for c becomes—

$$= \frac{c}{0{\cdot}622} = \frac{53{\cdot}18}{0{\cdot}622} = 85{\cdot}5 \text{ ft.-lb. ;}$$

and, therefore, $P \cdot V = 85{\cdot}5\ T_0$ for superheated steam (see p. 43), where T must be some temperature above that corresponding to the boiling-point of saturated steam at the pressure considered.

Other values for superheated steam are (see p. 274)—

$$Kp = 0{\cdot}48 \text{ b.t. units} = 374{\cdot}4 \text{ ft.-lb.}$$
$$Kv = 374{\cdot}4 - 85{\cdot}5 \quad = 288{\cdot}9 \text{ ft.-lb.}$$
$$y = \frac{Kp}{Kv} = 1{\cdot}296 \qquad = 1{\cdot}3 \text{ nearly.}$$

General Law of Expansion of Gases, when doing Work.— When any small increment of heat, dH, is supplied to a substance, there is a gain in intrinsic energy which may or may not be converted into work. In any case—

$$dH = Kv \cdot dt_0 + p \cdot dv.$$

When the expansion is adiabatic, no heat being received into the substance, the work done = change in intrinsic energy of the substance, and—

$$dH = 0, \text{ or zero.}$$

Therefore $$Kv \cdot dt_0 + p \cdot dv = 0 \quad . \quad . \quad . \quad (1)$$

For a perfect gas, $$p \cdot v = c \cdot t_0.$$

From which, $p \cdot dv + v \cdot dp = c \cdot dt_0$. But $c = Kp - Kv$;

therefore $$dt_0 = \frac{p \cdot dv + v \cdot dp}{Kp \cdot - Kv}.$$

Substituting this value in (1) for dt_0—

$$\frac{Kv}{Kp - Kv}(p \cdot dv + v \cdot dp) + p \cdot dv = 0.$$

That is, $$Kv \cdot v \cdot dp + Kp \cdot p \cdot dv = 0.$$

Divide by $Kv \cdot p \cdot v$, then $$\frac{dp}{p} + \frac{Kp \cdot dv}{Kv \cdot v} = 0.$$

Substituting y, and integrating—

$$y \cdot \log v + \log p = \text{constant ;}$$

from which, $$p \cdot v^y = \text{constant.}$$

With a certain mass of dry air contained behind a piston in a cylinder, the temperature remains constant if the piston be moved by some external agency and allows the air to expand without doing work. The energy contained in the mass of air undergoes no change —it is always equal to P . V ; thus following Boyle's Law and also that of Charles, because T does not change.

Again, if work be done while the piston moves, or say, that the pressure of air overcomes a resistance opposed to the piston, then energy in the form of heat is taken from the mass of the gas. The removal of this heat, and its conversion into mechanical energy or energy of motion, leaves the mass of the air at a lower temperature. The drop in temperature, if the pressure remain constant, would itself produce a less volume ; consequently, as the volume is allowed to increase also, the pressure at the end of the expansion is less than in the case where no work is done. The form of the expansion curve now follows that given by the expression, $P . V^n = \text{constant}$, in which n is always greater than unity (1) and generally less than 1·4.

Air Compression and Compressed-air Engines.—A simple graphical illustration of a cylinder and piston working under the stated conditions shows fairly clearly the result which may be expected when mechanical energy is converted into heat energy by compressing air

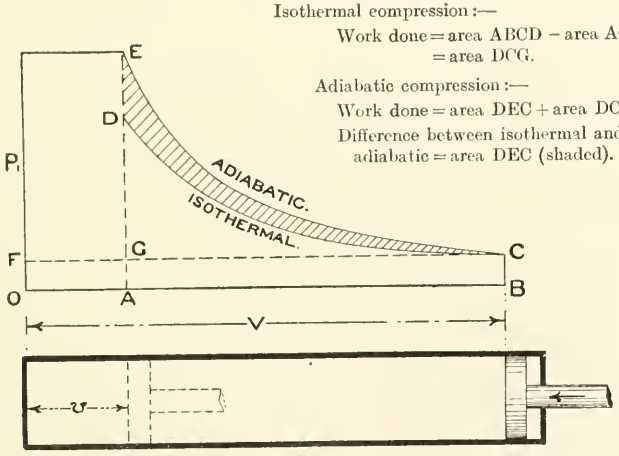


FIG. 133.—Work done during Compression of Dry Air.

(Fig. 133), or when heat energy is converted into mechanical energy by compressed air driving a piston before it and when the previous conversion of energy is reversed (Fig. 134).

In a submerged torpedo driven by compressed-air engines, *without supplementary heating*, the temperature of the working cylinders is kept nearly constant by the heat supplied from the surrounding water. Without this supply the engines might possibly be chilled and any moisture present in the air become frozen. Consequently, because the temperature of the air is higher than it would otherwise be at the end of the expansion, the pressure on the piston is better maintained during the expansion and more work is given out by the engine. In

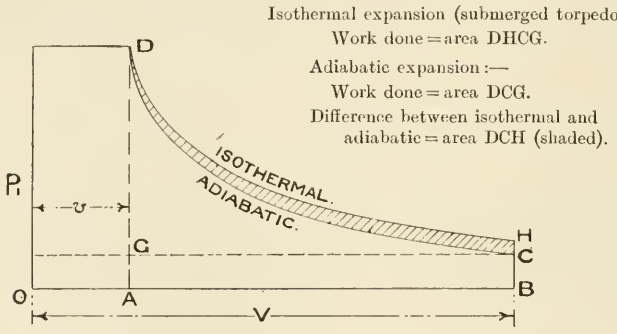


Fig. 134.—Work done during Expansion of Dry Air.

actual practice the importance of obtaining a temperature of the air in the torpedo storage-chamber as nearly as possible equal to that of its surroundings, is thus emphasised. If the temperature of the air be above that of the surrounding water, the pressure is decreased and the power decreased also. If the temperature be below that of the surrounding water (an almost impossible condition), the air in the chamber tends to expand and, being restricted in its volume, the pressure is increased (to perhaps a dangerous extent in an extreme case).

The highest practical efficiency of the machine in which air is compressed can only be obtained by a minimum rise in temperature of the air so compressed. This is fairly evident if we consider that any rise in temperature produces an equivalent increase in pressure which the machine has to overcome; and in doing so, more work must be done than if the temperature had remained approximately constant. Represented graphically, the extra work required is shown by the area contained between the two curves of the isothermal and the actual compression (Fig. 133).

In a purely supposititious case, when no cooling is effected either

by water cooling or radiation, etc., a large amount of the work done by the machine is transformed into heat and the temperature of the air is considerably raised. The rise in temperature is obtained as follows :

Let T_0 be the final temperature, V be the corresponding volume, and P be the pressure.

Let t_0 be the initial temperature, v be the corresponding volume, and p be the pressure.

And let $y = \frac{Kp}{Kv} = 1{\cdot}4$ for air.

Then from previous sections we have—

$$P \cdot V^y = p \cdot v^y \text{ and } \frac{p \cdot v^y}{P \cdot V^y} = 1.$$

Also $\frac{T_0}{t_0} = \frac{P \cdot V}{p \cdot v}$ approximately, air being nearly a perfect gas.

From which,

$$\frac{T_0}{t_0} = \frac{P \cdot V}{p \cdot v} \times \frac{p \cdot v^y}{P \cdot V^y} = \frac{v^{y-1}}{V^{y-1}} = \left(\frac{v}{V}\right)^{y-1}.$$

It is more convenient to measure the relative pressures, than the relative volumes, and therefore we proceed to get pressures in place of volumes.

$$P \propto \frac{1}{V^y} \text{ and } p \propto \frac{1}{v^y}.$$

Therefore $\frac{P}{p} = \left(\frac{v}{V}\right)^y$ and $\frac{v}{V} = \left(\frac{P}{p}\right)^{\frac{1}{y}}$.

From which, $\frac{T_0}{t_0} = \left(\frac{v}{V}\right)^{y-1} = \left(\frac{P}{p}\right)^{\frac{y-1}{y}} = \left(\frac{P}{p}\right)^{\frac{0{\cdot}4}{1{\cdot}4}} = \left(\frac{P}{p}\right)^{\frac{2}{7}}$.

As already pointed out above, such actual temperatures are seldom, if ever, reached in ordinary practice ; but an example of what might occur is instructive.

EXAMPLE.—A chamber is charged to 2250 lb. pressure per square inch, and the initial temperature of the air is 99° F. (=560 absolute F.) ; find the rise of temperature *if no cooling is allowed.*

$$\frac{T_0}{t_0} = \left(\frac{2250}{14{\cdot}7}\right)^{\frac{2}{7}} = (153)^{\frac{2}{7}}.$$

Taking logarithms—

$\text{Log } T_0 - \log t_0 = \frac{2}{7} \cdot \log 153$ — Log 153 = 2·1846914

$= 0{\cdot}6241975.$ — 2

7) 4·3693828

$T_0 = t_0 \times 4{\cdot}208$

$= 560 \times 4{\cdot}208$

$= 2356{\cdot}48.$ — Log 4·208 = 0·6241975

$T_0 - t_0 = 2356 - 560 = 1696°$ F. rise of temperature.

When air is compressed in an actual compressor only a moderate rise in temperature takes place, because there is always some radiation, etc., even when the pressure is pumped up very quickly ; in addition, water circulation is used to cool the air both during compression and between the stages of compression. In the matter of more efficient cooling great improvements have been made in recent years, and economy in power used for compression has been obtained in consequence.

Work done during the Isothermal Compression of Dry Air.—The compression being isothermal, the compression curve is hyperbolic and $V \propto \frac{1}{P}$, because T is constant. The problem comes under external work, and the work done, represented by the area DCG of Fig. 133, $= p_m(V - v)$.

EXAMPLE.—If 105 lb. weight of air, of absolute pressure 15lb. and temperature 102° F., be compressed isothermally in a chamber to 2400 lb. pressure per 1 square inch ; find the work done, and the horse-power usefully expended in air compression, if the time required to so compress it is 80 minutes.

The work done $= p_m \cdot (V - v)$, where $p_m = p_1 \times \frac{\log_e r}{r}$.

$p_1 = 2400 \times 144$ (lb. per 1 sq. foot).

$r = \frac{2400}{15} = 160.$

V = Volume of 105 lb. at 102° F. (= 563° abs. F.)

$= \frac{105 \times 12{\cdot}38 \times 563}{493}$ (1 lb. of dry air at 32° F. fills 12·38 cub. ft.)

$= 1485$ cubic feet.

v = Volume of same mass of air under 2400 lb. pressure per 1 sq. in.

$= \frac{1485 \times 15}{2400} = 10$ nearly.

$V - v = 1485 - 10 = 1475.$

Work done $= \frac{2400 \times 144 \times 1475 \times 4{\cdot}6}{160}$ ft.-lb. ($\text{Log}_e 160 = 4{\cdot}6052.$)

$$\text{Horse-power (for 1 minute)} = \frac{\text{work done}}{33{,}000 \times 80}$$
$$= \frac{2400 \times 144 \times 1475 \times 46}{33{,}000 \times 80 \times 1600}$$
$$= 5\tfrac{1}{2} \text{ about.}$$

This result is, of course, below the actual I.H.P. of the engine, because some rise of temperature takes place in each stage of compression of the pump, and work is done also in overcoming the friction of the working parts of the engine. In actual practice, the I.H.P. of the engine may be taken as about *four* times the horse-power as calculated above when assuming isothermal conditions.

Increasing the Thermal Efficiency of Torpedo Engines.—The power of a torpedo (compressed air) engine can be considerably increased by the addition of heat from some heating agent, or by burning some combustible substance and using the air charge to support combustion. The latter method supplies the more prolific means of augmenting the supply of energy, and is probably more capable of being safely used, because the rate of combustion can be more easily regulated to suit requirements.

With supplementary heating, the water usually surrounding the engine has an adverse effect on efficiency, because it takes away heat from the working substance instead of maintaining its temperature as in the simpler case of using compressed air only. This removal of heat can be only efficiently prevented by lagging the engine cylinders and other working parts exposed to the action of the cooling water, and might possibly be best attained in practice by excluding the water from the engine chamber, and by making the surrounding air serve as a non-conductor of heat from the engine to the water outside the torpedo.

Only a very rough comparison can be made, but it is known from petrol motor experiments that 105 lb. weight of air will support combustion for at least 7 lb. weight of petrol or spirit. In practice the charge of, say, 105 lb. weight of air compressed to 2400 lb. per square inch pressure develops about 50 horse-power for 1 minute, representing the energy of

$$50 \times 42{\cdot}3 = 2115 \text{ b.t. units.}$$

The available heat energy of 1 lb. of petrol = 16,500 b.t. units about; and if only one-half of this energy can be conveyed to the engine, there is still four times as much energy to be derived from it as can be derived from the air charge of the torpedo, when no heating is effected.

Steam Engines. Internal and External Work done during

Evaporation.—To illustrate the principle of internal and external work, an example may be taken of a vertical cylinder containing 1 lb., by weight, of water on which a piston rests and exerts a *constant pressure* (Fig. 135). Let the pressure on the piston be represented by P, the corresponding pressure per square inch by p, and the area of the piston on which the pressure is exerted by A; then—

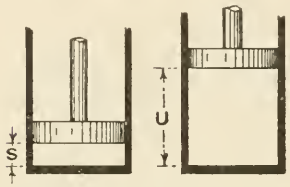


FIG. 135.—Work done by Steam.

$$P = p \,.\, A.$$

If all the water be evaporated under a pressure, p, then the piston will rise to a certain height; call this height u, in feet as measured from the bottom of the cylinder, and let s be the original depth of the water. Then the work done by the steam overcomes the resistance of the piston through a space, $u - s$, and therefore, during evaporation—

$$\text{External work done} = p \,.\, A \,.\, (u - s).$$

In this equation A . s is the volume occupied by 1 lb. of water and is the specific volume of the water (*i.e.* A . $s = 0{\cdot}016$). Again, A . u is the specific volume of the steam corresponding to a pressure, p, and is represented by v in the tables for saturated steam; so that—

$$\text{External work done} = p \,.\, (v - s) \text{ in foot-lb.; or}$$

$$= \frac{p \,.\, (v - s)}{J} \text{ in b.t. units.}$$

During the process of evaporation only the above quantity of heat, $\frac{p \,.\, (v - s)}{780}$, can be utilised in doing work, and the remainder, a very much larger quantity, enters the mass of the water and is used up in overcoming internal molecular cohesions and molecular variations of form and shape, *i.e.* resistances which are invisible and only manifested by the results. Therefore it may be stated that—

(*a*) *Internal latent heat* is that necessary to overcome the internal resistance of the water opposed to its conversion into steam. And—

(*b*) *External latent heat* is that necessary to raise the piston, and which, therefore, does no work in the mass of the water but is used on external bodies.

The sum of these two quantities represents the latent heat L of

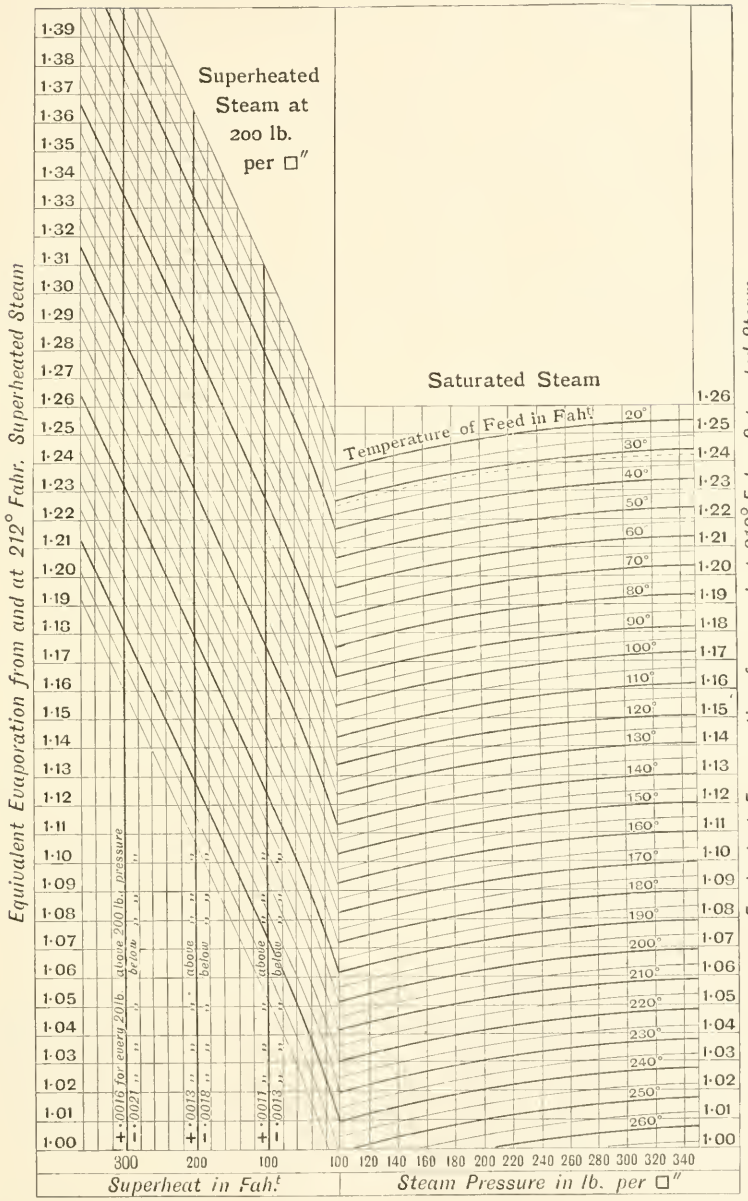


FIG. 136.—Factors for Equivalent Evaporation from and at 212° Fahr., for Saturated and Superheated Steam.

formation ; and denoting the internal latent heat by R, and expressing each part in b.t. units, then—

$$L = R + \frac{p \cdot (v - s)}{780}.$$

If the total heat H of formation of steam is required, then—

$$H = S + L = S + R + \frac{p \cdot (v - s)}{780}, \text{ where } S = \text{sensible heat.}$$

When $p = 315$, the value of v is about 88, and when $p = 14{\cdot}7$ (atmospheric pressure), $v = 1643$; therefore, approximately, $v = v - s$, and is approximately the same at all ordinary pressures.

And $$H = S + R + \frac{p \cdot v}{780}, \text{ approximately.}$$

If only a part, x, of the water is converted into saturated steam, and a fraction, $(1 - x)$, remains as water in suspension and mechanically mixed with the steam (as commonly found in practice), then—

$$H = S + x \cdot R + \frac{x \cdot p \cdot v}{780},$$

and the ratio between the internal and external latent heat does not change.

From the above considerations, it is evident that when steam is generated under constant pressure—as in a boiler, after steam has been raised to the working pressure—the measure of the external work done during evaporation is = the product of the pressure and the change in volume (from water to steam at the working pressure) ; and adopting the proper units for saturated steam, this product approximately $= p \cdot v =$ external work done during evaporation at constant pressure.

Any further increment of work obtained from the steam is done at the expense of the internal energy, contained in the steam in the form of heat, and in practice a relatively large quantity of work is so obtained by utilising the expansive quality of steam. It is evident that the work done *during admission* is practically constant for the same weight of steam at all pressures, and when the expansion curve of pressures is assumed to follow the hyperbolic law ; then—

Total work done = work during admission + work during expansion

$$= p \cdot v + p \cdot v \cdot \log_e r,$$

the increment is expressed by the last quantity. This increment increases with r, but in a rapidly decreasing ratio within the practicable limits of ordinary pressures and temperatures.

The action of steam expanding in a cylinder (Fig. 137) behind a piston may now be considered :

1. When the piston is moved by some external force so that the steam (OD), *originally saturated*, expands and no external work is done, the quantity of heat energy contained in the steam remains unchanged if we neglect radiation, conduction, and other sources of loss. The pressure becomes less by expansion, as the steam has more space to fill, and the total heat necessary to maintain the steam in a state of saturation is reduced (see table, page 18) ; consequently a small quantity of heat is released from its duty in maintaining the state of saturation, and the steam becomes superheated (curve D1), *i.e.* it is raised to a temperature above that necessary to keep it in a state of saturation. The specific heat of superheated steam near the saturation temperature is about 0·48, and therefore for each b.t. unit set free per 1 lb. weight of steam the temperature of each 1 lb. will be raised 2·083° F. above the temperature of saturation ; steam under these conditions being considered to act almost exactly as a gas.

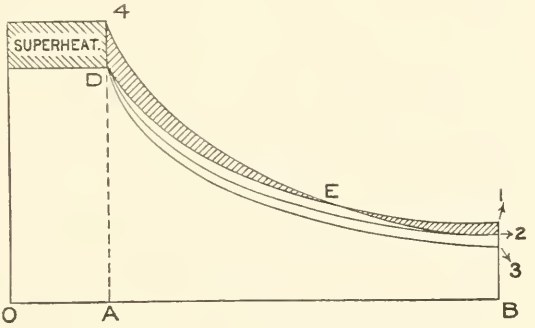


FIG. 137.—Expansion of Steam under Various Conditions.

2. When the piston is moved by some external force such that the steam (OD), *originally saturated*, expands and does some external work exactly equivalent to the heat set free by expansion or that used to superheat it when no external work is done, then the quantity of heat remaining in the steam is just sufficient to keep the steam in a state of saturation (curve D2). The same condition may be obtained in practice—but is unusual—either when the steam, in addition to doing none or a small quantity of work, gives out heat by radiation, etc., or when the steam does work on the piston and receives heat from an external source, such as the jacket or other

source, in sufficient quantity to maintain the steam in a state of saturation.

3. When the steam (OD), *originally saturated*, expands and does work, as in the ordinary steam engine when unjacketed or jacketed, heat is abstracted from the steam and transferred into work done on the piston. In this case, generally the steam at the end of the expansion contains an insufficient quantity of heat to maintain it in a state of saturation ; condensation therefore takes place and the pressure falls, in addition to the fall caused by expansion, and the steam becomes wet (curve D3). In the ordinary stage-expansion engine there is usually a drop in pressure when the steam enters the receiver between the stages, during which drop expansion takes place and no work is done (as in case 1 above), and some heat is therefore set free which either tends to re-evaporate any moisture which may be present in the steam, which is the usual condition, or tends to superheat the steam if already saturated, which is a very unusual condition.

4. When the steam (O4), *originally superheated*, expands and does work on the piston, as in steam engines using superheated steam, heat is abstracted from the steam and transferred into work done. The steam remains superheated (curve 41) at the end of the expansion if the work done, expressed in b.t. units, is not greater than $(T - t) \cdot K_p$— where T = Temperature of the superheated steam on supply to engine ;

t = Temperature of same steam when in state of saturation ;

K_p = Mean specific heat of superheated steam at constant pressure, between the temperatures T and t. (Fig. 132B.)

If the work done is only slightly in excess of this quantity, the steam at the end of the expansion (curve 42) may be in a state of saturation, *i.e.* when the work done, in b.t units $= (T - t) \cdot K_p + (H - h)$, where—

H = Total heat of formation of steam at the initial pressure, and at the original temperature t ;

h = Total heat of formation of steam at the final pressure, produced by expansion.

If the work done is in excess of the quantity $(T - t) \cdot K_p + (H - h)$, then the steam becomes wet, as the quantity of heat contained in it at the end of the expansion is less than h and insufficient to keep it in a state of saturation.

In many three-stage expansion engines which may be supplied with steam considerably superheated, the steam remains superheated during the first stage (or H.P. cylinder) and a moderate amount of

superheat is carried into the second stage (or M.P. cylinder); but during the expansion in the latter stage the steam passes from the superheated, through the saturated, into the wet state. In the third stage (or L.P. cylinder) the steam is generally wet throughout, notwithstanding partial re-evaporation, or tendency to superheat, obtained by the drop in pressure in the L.P. receiver and by any heat which may be received into the steam through the medium of the cylinder steam jacket.

The possible practical value of superheat in steam engines is much greater than might appear from the above considerations. In the first place, the maximum efficiency, considered as a perfect heat engine, is increased by the increase in temperature from T to T_s, because in place of an efficiency, $\frac{T-t}{T+461}$, for saturated steam, an efficiency of $\frac{T_s-t}{T_s+461}$ is obtained for superheated steam.

In the second place, superheat prevents initial condensation of the steam in one or more cylinders, and dry steam, especially when superheated, being a very poor conductor of heat, allows only a very small quantity of heat to escape through the medium of the cylinder walls. Steam when not superheated, or slightly moist, as it is usually when entering any cylinder, readily condenses from the fall in temperature caused by the comparatively cool cylinder; small globules of water are formed in the body of the steam, and, being deposited on the cylinder walls, rapidly convey heat away, thus occasioning a loss which does not occur when the steam remains superheated. As the cylinder condensation loss is less when superheat is used, a greater quantity of work is obtained from each 1 lb. weight of steam used in the engine. A greater quantity of heat is necessarily supplied to the steam, but the greater proportion of useful work obtained from superheated steam increases the efficiency by economising the heat supplied by the fuel.

Action of Steam Expanding in a Convergent-divergent Nozzle.—In " impulse " turbines the steam is allowed to expand in a nozzle adiabatically (*i.e.* without either taking in heat from or giving heat out to external bodies); work is done by the steam in promoting motion in itself by reason of a fall in pressure and a corresponding fall in temperature. The greater the fall in pressure and temperature, the greater becomes the acceleration of velocity and energy of motion, or kinetic energy.

The nozzle may be any length, either very short (as in the De Laval turbine, only 2 or 3 inches) or very long (as in turbines of the Parsons type, where the nozzle extends for the whole axial length of the blading). The nozzle is usually arranged with a rounded entrance, allowing the steam to converge to the throat in which the pressure is never lower than 0·58 of the initial pressure. From the throat to the discharge end the nozzle area expands until it corresponds to the proper area of discharge for the final pressure. In calculating the discharge area, it should be clearly understood that work is being done by the steam inside the nozzle in creating motion of itself, and that, therefore, heat is being converted into the kinetic energy of motion. If the steam be originally dry saturated, some of it must necessarily be condensed to water during the expansion.

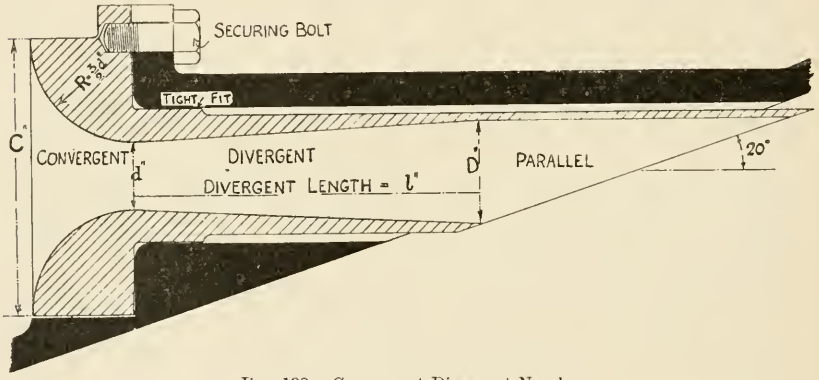


Fig. 138.—Convergent Divergent Nozzle.

Fig. 138 represents a convergent-divergent nozzle, and the subject is dealt with in Chapter XX.

Some experiments made by the author, in connection with the Dickson-Corthesy turbine during 1913–14, tend to show that a good efficiency of conversion is obtained with circular nozzles proportioned on the following basis—

d = Minimum diameter (neck ; or vena contracta) ;

C = Diameter of entrance = $4d$;

L = Length of entry from C to $d = \frac{3}{4}$ diameter at d ;

R = Radius of curvature of entry = $\frac{3}{4}$ diameter at d.

(The centre of curvature lies in the diameter (produced) of the neck d, and the curve extends from C to d, where it merges into the bore.)

From the neck d, the bore diverges at an angle of 10 degrees until the maximum diameter D corresponds with final pressure conditions. From this point the nozzle bore is parallel for a length necessary to allow for part to be flushed off to an angle corresponding with that at which the nozzle is fitted in the stator (generally 20 degrees) as shown in Figs. 216 and 222.

In practice the diameter at C is considerably less than $4d$, but it should never be less than $2d$. For convenience the fitting part can be made conical, but some security must be fitted to prevent the nozzle turning on its seating, and for fixing its position correctly at the discharge end. The parallel discharge is necessary for direction of flow, and to prevent spreading and waste of energy from turbulence. The space around the discharge end of the nozzle is provided to allow free expansion, and to reduce loss of heat energy by conduction to the casing.

PART V

THE MARINE STEAM ENGINE (PISTON TYPE)

CHAPTER XVI

VERTICAL ENGINE AND FRAMING

Vertical Engine and its Advantages.—Two types of marine engines are now in general use; turbines, described in Part VI., and the vertical-inverted direct-acting reciprocating engine, commonly known as the Vertical Engine, which is described and illustrated in this and the following chapters. The Internal Combustion Engine is also generally a vertical direct-acting reciprocating engine, and is described and illustrated in Part XI. at the end of this book.

The vertical steam engine has some advantages over previous types, which were horizontally arranged as direct-acting, trunk, and return connecting rod, engines. The advantages are as follows:

(1) The ground space occupied by it is much less;

(2) Greater accessibility for erection and repair;

(3) Less friction and wear on all working parts and especially the reciprocating pieces of the mechanism;

(4) A longer stroke could generally be used and consequently a slower rate of revolution which contributed to greater propeller efficiency and other advantages, especially less steam leakage, better balancing, and freedom from vibration. These advantages more than balanced the increased weight and space involved in a longer stroke.

Necessity for a Guide.—In Fig. 139 diagrams are shown of a vertical engine in the relative positions for the up and down strokes. When the engine is going "ahead" the pressure on the guide G is in the *same direction*, whether the piston D is either pushing the rods E and R downward or pulling the rods upward; and so long as the

shaft continues to move in the same direction of rotation *under the influence of the piston*, only the one guide shown is necessary to keep the piston rod in line with the axis of the cylinder *A* and piston *D*.

If the direction of motion of the shaft be reversed, to go "astern," a guide is necessary on the opposite side of the crosshead to take the pressure in the opposite direction. This arrangement is fitted in many mercantile engines; but in others a lip is formed in connection with the guide used for going ahead, and embraces projections on the crosshead so as to take the astern pressure on the guide in the opposite direction. This arrangement is clearly shown in Figs. 142, 165, 166, and 167.

The *direction of rotation* of the shafts when going "ahead" varies in a twin-screw ship correspondingly with "inward" and "outward" turning screws. For inward turning, fitted in many naval vessels between 1895 and 1902, the *port* shaft revolved as shown in Fig. 139, when going ahead as viewed from aft, while the starboard shaft revolved in the opposite direction; the cranks when above the shaft centre turn *towards* each other. With outward turning screws, the cranks when going ahead and above the shaft centre turn *away* from each other; and the back frames, to which the guides are attached, are back to back with the mid-line bulkhead between them. (See also Chapter XXVIII. "Inward and Outward Turning Screws.")

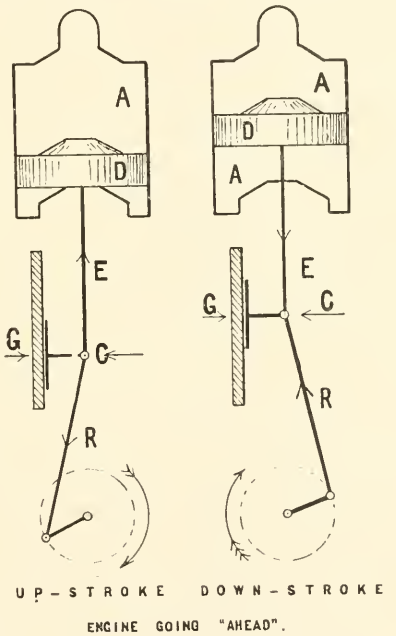


FIG. 139.—Necessity for a Guide.

Obliquity of the Connecting Rod.—It should be noticed that when the piston is at mid-stroke *M*, the crank is not exactly at right angles to the line of stroke (produced). In Fig. 140, *A* and *B* are the top and bottom ends of the stroke of the crosshead, which exactly agrees with the relative position of the piston in its stroke because they are rigidly connected. The length of the connecting rod = $OM = AN = BK = MR$, as shown in the figure. When the crank is at *OP*,

at right angles to AO, the position of the crosshead is at C, where $PC = MO$, and C is not coincident with M but a little below it. When the crosshead is at mid-stroke M, the crank is at R (or in a similar position on the opposite side of AO), and the position of the crank relative to AO is at L, where LR is at right angles to AO. If the rod is of infinite length, then C becomes coincident with M, and L with O, and the relative motion is *harmonic*.

The error, so called, due to obliquity is generally neglected in

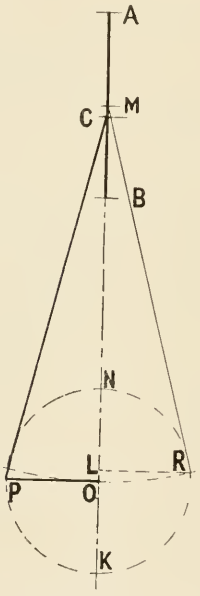


Fig. 140.—Obliquity of Connecting Rod.

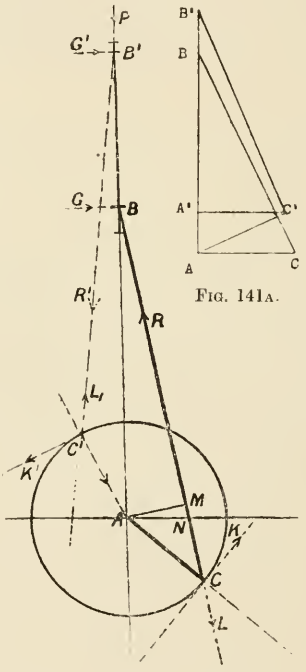


Fig. 141A.

Fig. 141.—Pressure on Guide.

elementary calculations relative to the connecting rod, but is of importance in the accurate design and setting of the slide valve, as well as in closely approximate calculations of the various forces and turning moments acting on the crank.

Pressure on the Guide.—In considering the motion of the ordinary reciprocating engine, the various forces and resistances acting in various directions must also be included. In Fig. 141 a diagram is given of the forces with their directions. The full lines show any position of the crank AC, and the connecting rod BC, on

the down stroke. Suppose the total force, or pressure, exerted on the piston to be P, then this force P acts in a line, B′B. At B part of this force is transferred to the connecting rod, and to alter the direction of this force into the direction of the connecting rod for transference to the crank, a variable force, or resistance, G, is necessary. G is the resistance of the guide, and acts in the direction shown by the arrow, that is, perpendicularly to the guide surface.

At the point B we have three forces, which may be momentarily considered in equilibrium. Denote the angle CBA by θ. Then—

$$G = P \,.\, \tan\, \theta,$$

and at the top and bottom of the stroke (*i.e.* the dead points) the angle vanishes ; consequently—

$$\tan\, \theta = \text{zero}.$$

Therefore G = zero at these points, and G will be a *maximum* when tan θ is a maximum ; that is, when θ is a maximum. A little consideration shows that θ is greatest when the angle BAC is a right angle.

Proof (Fig. 141A).—In the triangles *BAC* and *B′A′C′*, *A′C′* is drawn perpendicular to *AB*. *BC* is equal to *B′C′*, the length of the connecting rod ; *AC′* is equal to *AC*, the length of the crank ; and each is greater than *A′C′*.

Therefore the angle *ABC*, which is subtended by *AC*, is greater than the angle *AB′C′*, which is subtended by *A′C′*. *Q.E.D.*

Area of Guide Surface.—The area of rubbing surface on the slipper on the crosshead at the end of the piston rod must be made of a suitable area for the maximum pressure on the guide, or G. For fast-running engines, an area which limits the maximum pressure when going " ahead " to about 40 lb. per square inch of area is allowed.

Turning Moment or Crank Effort.—Let the crank, when in any position, AC, make an angle CAB equal to x with the vertical line, and let AC equal b, and BC equal a ; draw AM perpendicular to BC. Then if P be acting downwards at B—

$$\text{The turning moment at C} = R \times AM$$
$$= R \times b \times \sin\, (\theta + x).$$

$$\text{But R, the force acting on the crank at the point C} = \frac{P}{\cos\, \theta}.$$

$$\text{Therefore the turning moment} = \frac{P \,.\, b \,.\, \sin\, (\theta + x)}{\cos\, \theta};$$

and as b and a can be measured, the angles θ and x can be calculated.

Pressure allowed on the Rubbing Surfaces of Bearings.—There is an enormous difference in the pressures allowed per square inch of rubbing surface in the various bearings, even when the lubrication is nearly as efficient as it can be made ; and, as an object-lesson in proportion, some typical figures are given below, with some of the principal reasons for their existence.

For bearings, such as the cross-head, crank-head, and shaft bearings, the effective area on which the pressure is exerted is considered to be that of the " diameter of the pin multiplied by its length," and a little consideration shows that this is very nearly the correct theoretical amount.

Maximum pressures allowed per square inch of effective surface :—

Connecting rod, top-end bearing . . .	1200 to 1250 lb.
There is only an oscillating motion on this bearing, not a complete rotation, with a very low velocity of rubbing.	
Connecting rod, crank-end bearing .	500 ,, 550 ,,
There is a complete rotation, one revolution to each revolution of the engine, but with a varying velocity.	
Guide surface for ahead working . .	75 ,, 140 ,,
This small amount is due to difficulty in retaining the lubricant on the rubbing surfaces.	
Main bearings	300 ,, 350 ,,
Thrust rings, effective surface only (old type)	40 ,, 70 ,,
This small amount is due to difficulty in retaining the lubricant, which is easily thrown off by the centrifugal action.	
Michell Thrust Block	300 ,, 400 ,,

The Engine Framework.—In Fig. 142, and in the reference table below, the various names and positions of the principal parts of a vertical inverted engine are stated. The figure shows only one (the M.P.) cylinder of a three-stage expansion engine, but the arrangement is similar for every cylinder ; and the whole engine with its three or more cylinders has a common seating, with sole plates joined together so as to form one complete framing.

For the reception of the engines, the ship's framing which is to support them is specially built up on the inner bottom if a double bottom is fitted, or on the inner frames if the ship is not so fitted. Double bottoms are fitted in all important ships under the machinery and boiler spaces at least, and generally in naval vessels of over 2500 tons displacement. In mercantile vessels the machinery and boilers

are generally secured to the *tank tops*, which correspond to an inner bottom fitted about horizontally above the outer bottom.

Engine seating is the name generally given to the framework *K* and *N*, which is built in by the shipbuilders and riveted to the ship. The transverse frames *K* are rigidly connected, by riveting, with the box frames *N*, running longitudinally the whole length of the crank shaft. It is good practice to make the thrust-block seating continuous with the engine seating; this distributes the thrust over a considerable number of frames of the ship. The engine seating should be unyielding in shape at all times, so that the shafting should not be subject to varying alignment in its bearings, which are supported by the engine seating.

The *sole plate or bed plate* is fitted above the seating. It is a complete structure in itself, made up of the sole plate (*S* in figure) and the beams *H*, which join up the ends of the sole plates longitudinally. Proper intervals and spaces must be left for the working of the cranks and eccentrics in both the sole plating and in the engine seating (see also Fig. 172). If possible, the sole plate is made of forged steel; but generally it is too complicated, and a casting is substituted. The plates are *bolted* down to the engine seating, and solid liners of cast iron are fitted (shown in cross section in the figure) between the sole plate and seating; the liners are trimmed as necessary to produce correct alignment of the bearings, in which the shafting rotates. The beams *H* are bolted to *S*, but in some cases rivets are used. Recesses are formed in the sole plates for the reception of the main bearings. Every precaution is taken to secure a rigid structure as a support for the engine *as a whole*, so that when subject to any stress it shall move as one piece and bring the least possible stress on any individual part. Non-rigidity of the framing supporting the shafting produces a varying alignment, and has frequently resulted in heated bearings and fracture of the shaft and its connections.

Nearly upright standards or *columns*, *F* and *B*, are fitted above the sole plates, and bolted down to them; nuts and check nuts are fitted on the securing bolts, and in addition, the ends of the bolts are generally riveted over to prevent any possibility of the nuts slacking back. All bolt fastenings used about the framing and columns are usually secured in the same way.

In some instances, and particularly for the ordinary cargo steamer, the front standards *F* are made in one piece, and identical with the back column *B*. Twin gudgeon pins are frequently fitted with this

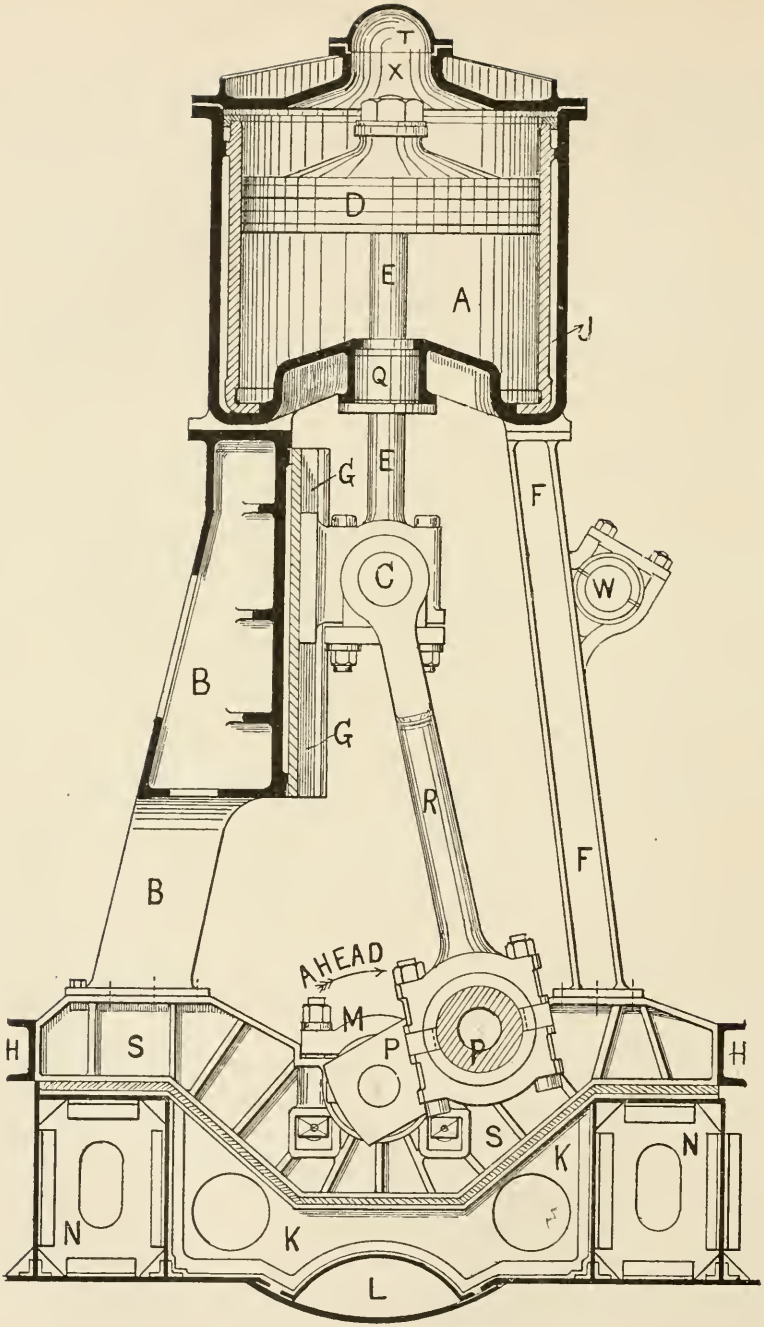


Fig. 142.—Parts of a Vertical Engine.

arrangement. *Double guides*, as above described, are particularly useful when the engines are much used for astern working, such as river steamers and tugs, because the areas of the rubbing surfaces of the slipper and guide are equal for both directions of rotation of the screw. Many engines are not worked astern for long periods, the extra astern guide surface is not required, and double guides are not fitted. In addition, there is some increase in weight, and the working parts are perhaps less accessible than with the *open-fronted* framework, as shown in Fig. 142.

REFERENCES TO FIG. 142

A, Cylinder.
B, Cast-iron back frame or column.
C, Gudgeon pin at top end of connecting rod ; and slipper.
D, Piston.
E, Piston rod.
F, Front columns, cast iron or forged steel.
G, Guide, or shoe ; for crosshead, or gudgeon, slipper.
H, Steel beams, joining up the sole plates *S* longitudinally, and in small engines made in one piece with the sole plates.
J, Jacket steam space.
K, Engine seating, made of steel plates and frames riveted to inner bottom or transverse framing of the ship.
L, Limber hole for draining crank pits to bilge suction.
M, Main bearing for crank shaft, fitted in recess in sole plate.
N, Engine seating, made of steel plates and frames in box-shape, and connecting up the frames *K* longitudinally.
P, Crank and bottom end of connecting rod.
Q, Packing gland.
R, Connecting rod.
S, Sole plate, steel casting or forging, forms recess for main bearing.
T, Manhole cover.
W, Bearing ; for weigh shaft used for reversing and controlling the link motion.
X, Cylinder cover.

Chocks and Ties.—The cylinders of large engines are generally secured together by tie rods, or parts of them are flanged and bolted together. The system, generally advocated for large engines, of fitting each cylinder and its supports as a separate structure, appears to be going out of fashion ; this separation allowed for difference of

expansion of the cylinders at top and the shaft bearings below, and thus tended to produce good alignment at all times of the rods and cranks. Although the engines are larger, and the working temperatures higher, no harm appears to have been caused by rigidly connecting the cylinders together. A good example of this is shown in Fig. 145, where the four cylinders, side by side, are bolted together. Many four-cylinder engines are secured in pairs by flange joints, and each pair is secured to the other by tie rods between the joints, as shown in Figs. 303 and 304.

All naval engines are fitted with ramming chocks, to relieve the working parts of the engine of the shock caused by collision or ramming, and thus prevent fracture of the machinery, in addition to the ordinary tie rods. These are not rigid connections, but arranged to come into play only when the engine is slightly displaced. When the usual side-by-side engines are fitted, the upper parts of the cylinders are sometimes stayed to the mid-line bulkhead between the two engine rooms. This arrangement adds considerably to the rigidity of the structure when the ship is heeled or rolling.

Height of Engine.—The height of an engine, which is measured from the top of the cylinder cover to the centre of the crank shaft, depends practically on the length of stroke adopted. For general purposes the connecting rod is made twice the length of stroke, and the distance between the centres of the crank and shaft is evidently always equal to half the stroke. Putting the crank on the top centre, the height of engine = stroke + length of connecting rod + throw of crank + thickness of piston and cylinder clearance at top and bottom + height of cylinder cover + length of stuffing box and clearance allowed between it and centre of crosshead

$$= 1 + 2 + \tfrac{1}{2} + \tfrac{3}{4} + \tfrac{1}{2} + \tfrac{3}{4}, \text{ in strokes;}$$
$$= 5\tfrac{1}{2} \text{ strokes, more or less.}$$

For an engine of $1\frac{1}{2}$ feet stroke, the height $= 1\frac{1}{2} \times 5\frac{1}{2} = 8'\ 4''$ about.
,, ,, of 4 ,, ,, $= 4 \times 5\frac{1}{2} = 22$ feet.
,, *Caronia* of 5′ 6″ stroke ,, $= 5\frac{1}{2} \times 5\frac{1}{2} = 30$ feet.

(The length of connecting rod is rather over twice the length of stroke in this instance.)

Height, Space, and Weight compared.—In the formula for calculating the I.H.P. developed in any cylinder whose piston area is A, it is evident that an increase of I.H.P. can be obtained by increasing P, the mean pressure. P may be increased in two ways, either

by increasing the boiler pressure and retaining the same cut-off and ratio of expansion, or by retaining the same boiler pressure and using a later cut-off. Both these methods were adopted in naval practice to save space directly and weight indirectly; the highest pressures (up to 270 lb. per square inch) at the engines and a late cut-off (generally at $\frac{3}{4}$ stroke at full power) being provided for.

If P and A are limited, an increased power can be obtained by increasing the piston speed ($=2$ L.N.). There was a gradual increase in this direction, until the piston speed for battleships became about 960 feet per minute, and in destroyers and small craft about 1200 feet. The same piston speed can be kept by increasing N and decreasing L, while still retaining the same I.H.P. This decrease of stroke L reduces the height of the engine and, consequently, the space occupied by it, and also its weight. The increased number of revolutions decreases the size of shafting necessary (see Chapter XVIII. for formula for diameter of shaft), and also of its connections; this also contributes to reduced weight and space. The above conclusions explain why the same I.H.P. was obtainable with considerably less weight and space in a destroyer whose engines moved at a rate of 400 revolutions per minute than in a battleship whose engines made 120 revolutions per minute. For the same reasons the engines of a large mail steamer, say the *Caronia*, with 5′ 6″ stroke and 100 revolutions per minute, are heavier and occupy more space than those of an engine of 4 feet stroke, although the factor of safety, or margin of strength, allowed may be the same.

Higher pressures involve stronger engines, with stronger attachments and working parts, so that little economy in weight is effected directly from their use. But indirectly, an economy of fuel is obtained by using a greater rate of expansion with the higher pressure, so that a somewhat smaller engine is necessary for the same I.H.P.

Economy of weight and space are effected in reciprocating machinery by—

1. Increased pressures, producing greater mean pressures on the pistons.
2. Increased speed of piston and revolution.
3. Better materials, allowing greater working stress for the weight employed.
4. Using greater rates of combustion in the boiler furnaces.
5. By using water-tube boilers, particularly those with upright tubes.

Number and Arrangement of Cylinders.—The number of cylinders has been gradually increased to four where turbines have not been adopted. Very large cylinders require much staying, and the piston rod must also be proportionately large to withstand the possible bending stresses which are put on it. Although a rod may be quite

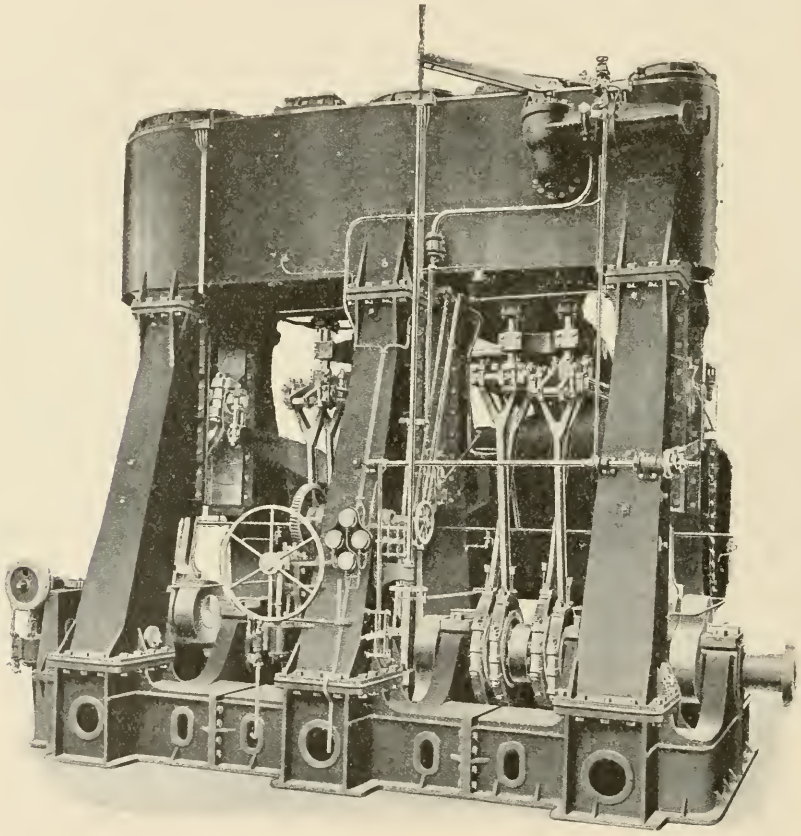


Fig. 143.—Engines of s.s. *North Western Midler.*

strong enough, it must be also made *stiff* enough for its work; this remark is also true particularly of large pistons. In the merchant service, where height is not of so much consequence, the increase in number of cylinders does not necessarily increase the number of cranks on the shafting, as the cylinders can be arranged tandem, one above the other, with a piston rod common to two cylinders. This arrange-

ment has been followed in a few cases where alterations have been made to introduce a greater number of stages of expansion to engines already fitted in ships. The present usual arrangement is to fit the cylinders *side by side*, each cylinder having its connecting rod attached to its own crank.

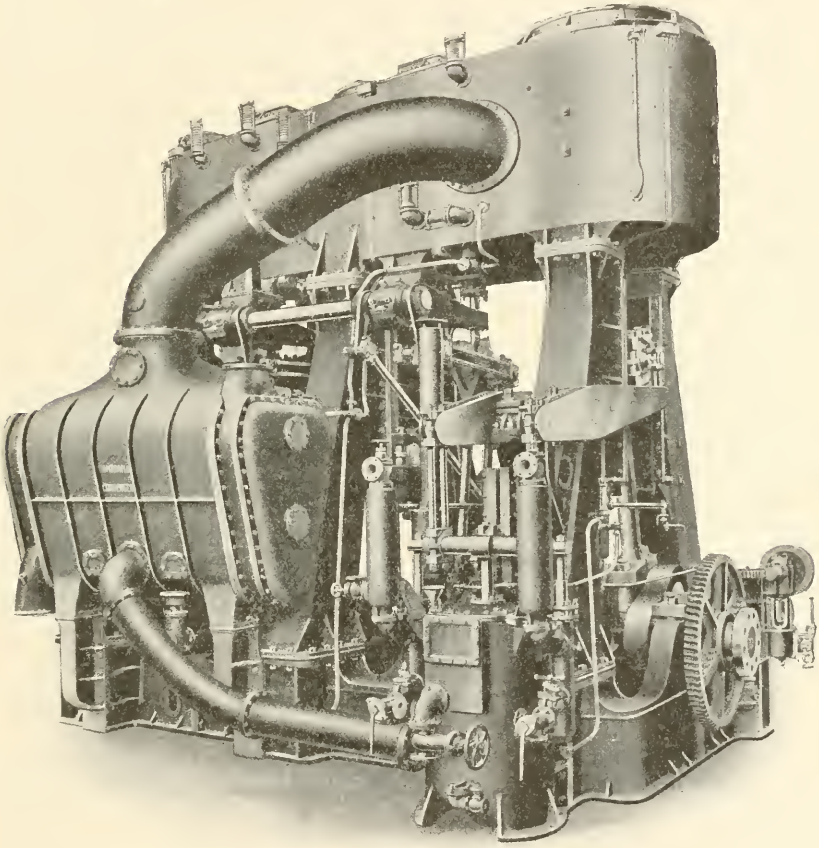


FIG. 144.—Three-stage Expansion Engines, s.s. *North Western Miller*.

Three-cylinder, Three-stage Expansion Engines. — The machinery of the s.s. *North Western Miller* was supplied by Messrs. Richardson, Westgarth & Co., in accord with the requirements of the late Sir Stephen Furness for the Furness-Withy line of steamships, and is illustrated in Figs. 143 and 144.

The engine is of the three-cylinder, three-stage expansion type,

with cylinder diameters of 29, 49, and 80 inches, with a stroke of 54 inches. The working pressure is 180 lb. per square inch, supplied by three boilers of the mercantile cylindrical single-ended type, 17′ 6″ diameter by 12 feet long, with suspension furnaces. The H.P. slide valve is of the cylindrical type, and the M.P. and L.P. are of the slide valve type. The M.P. valve is balanced for relief or equilibrium on the Martin and Andrews principle, and the weight of the L.P. slide valve is taken by a balance piston and cylinder.

The engine framing forms double guides for the piston rod ends, as usual in mercantile practice, and the crossheads are of the twin gudgeon type, as shown in Fig. 167.

The air pump is of the Edwards type, Fig. 321, and worked by rocking levers from the L.P. crosshead. The air pump discharges the feed water into a Cascade gravitation filter, Fig. 335, and then it passes to a pair of independent feed pumps which discharge the water through a Compensating de-aerating surface feed heater into the boilers.

The Contraflo system of condenser and accessories is fitted, including exhaust steam feed heating. In addition, for use in port, an auxiliary condenser, Fig. 325, is fitted with a Cascade filter, Compensator heater and float controlled feed pump.

Steam and hand starting gear is fitted, and there is also a small turning engine, seen on the right of Fig. 144.

Four-cylinder, Three-stage Expansion Engines.—In many cases the large L.P. cylinder has given place to two smaller ones of similar total capacity. The two L.P. cylinders make the engine rather longer, the frictional loss is greater with four cranks and connections than with three, and there is also a slight increase in weight; these are the disadvantages. The advantages are :—

1. Better balancing of the engines as a whole, Fig. 149, producing freedom from vibration with fast-running engines, and generally in all modern ships.

2. The great increase in the size of L.P. cylinders required for large powers makes it extremely difficult to construct cylinders and pistons for safe working; the exceedingly rapid reciprocation of the piston enormously increases its momentum, and there is consequently less difficulty in producing smaller pistons of sufficient stiffness and strength.

Each of the two L.P. cylinders is made of one-half the volume of the single cylinder otherwise required, but each, with its connections, is made strong enough to develop about five-eighths of the power

developed in each of the H.P. and M.P. cylinders. This increase of strength is particularly convenient with the closed exhaust system, where some of the steam from the auxiliary exhaust system is made to do work in the L.P. cylinders before passing into the condenser. A plan of an arrangement of cylinders, etc., is shown in Fig. 145.

Four-stage Expansion.—The naval engine is a compromise which embraces the greatest possible power on the least possible weight, in the least possible space, with a moderate economy at full power, and a relatively low fuel consumption at cruising speeds. As an example of the increased weight of machinery involved in mercantile practice, that of the s.s. *Saxonia* is actually double that of

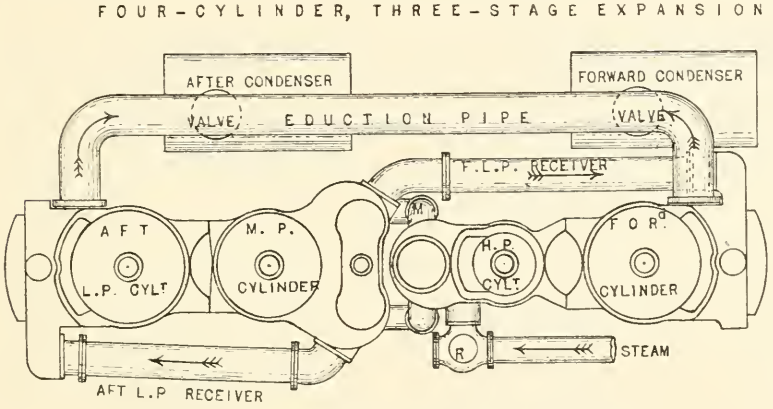


FIG. 145.—Four-cylinder, Three-stage Expansion Engine.

H.M.S. *Minerva* or *Hyacinth*, and the power is slightly less, although possibly more easily maintained.

Ships of the mercantile marine are built to steam at some regular speed, which is obtained by exerting the engines at very nearly their full power; economy of fuel, and not economy of weight and space, is the primary object. The extra weight and space under these conditions is soon saved in economy of fuel.

The twin engines of the Orient-Pacific s.s. *Orontes* are a good example of mercantile practice with four-stage expansion. The working pressure is 215 lb. per square inch. The engines are of the usual side-by-side arrangement. The H.P. cylinder is placed forward, next abaft is the L.P., then the 2nd M.P., and abaft all the 1st M.P. cylinder. The engines are balanced on the principle of varying crank angles, obtained from the calculation of inertia forces and couples

explained later in connection with Fig. 149 ; the H.P. crank leads the 1st M.P. crank by about 80°, followed by the 2nd M.P. at about 90°, which is about 100° in advance of the L.P. crank, leaving about 90°

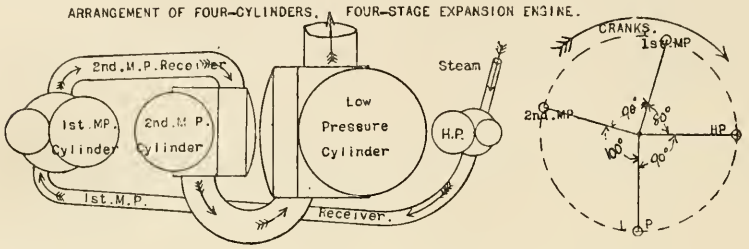


Fig. 146.—Four-stage Expansion Engine.

between the last and first. The crank angles and course of the steam are shown in Fig. 146.

S.S. "Agamemnon."—The general arrangement of one of the

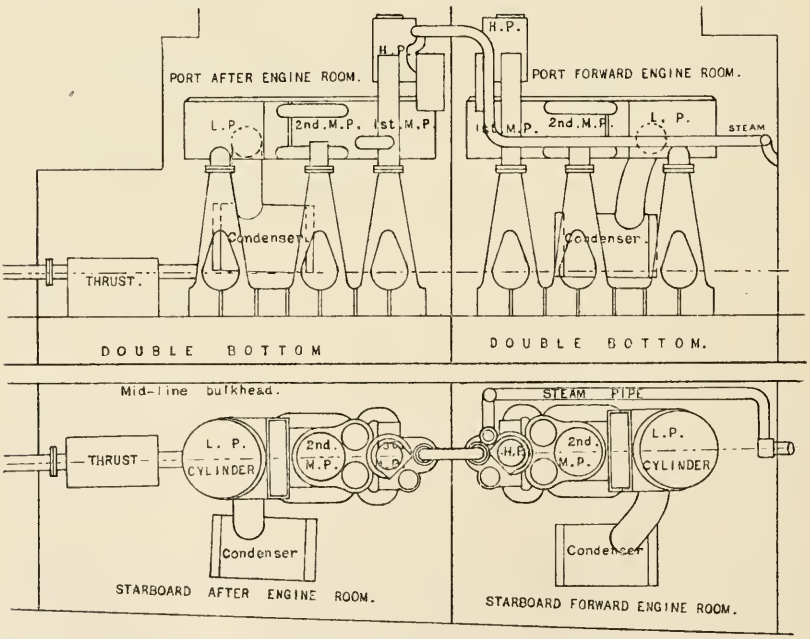


Fig. 147.—Engines of s.s. *Agamemnon.*

eight-cylinder engines of this twin-screw steamship is shown in Fig. 147. There are two engine rooms to each shaft, and four cylinders in each.

Dimensions, etc., of Notable Atlantic Steamers

Steamer's Name.	Builders.	Date.	Moulded Dimensions.			Proportion of Length.		Draught.	Displacement.	Gross Tonnage.	Cylinders.		Boilers.			Indicated Horse-Power.	Speed on Trial.
			Length.	Breadth.	Depth.	To Beam.	To Depth.				Diameter in Inches.	Stroke.	Heating Surface.	Grate Area.	Working Pressure.		
			ft.	ft. in.	ft. in.			ft. in.	tons.			in.	sq. ft.	sq. ft.	lb.		knots.
Great Eastern .	Scott Russell . .	1858	680	83 0	57 6	8·192	11·826	25 6	27,000	24,360	Screw, four 84-in.; paddle, four 74-in.	S., 48-; P., 17-	..	..	30	7,650	14·5
Britannic . .	Harland & Wolff .	1874	455	45 0	36 0	10·111	12·640	23 6	8,500	5,004	Two 48-in., two 83-in.	60	..	..	70	5,500	16
Arizona . .	Fairfield Co. .	1879	450	45 2	37 6	9·955	12·000	22 0	..	5,147	One 62-in., two 90-in.	66	..	..	90	6,300	17
Servia . .	Clydebank Works .	1881	515	52 0	40 6	9·903	12·716	23 3	9,900	7,392	One 72-in., two 100-in.	78	27,483	1014	90	10,300	17
Alaska . .	Fairfield Co. .	1881	500	50 0	39 8	10·00	12·607	22 0	..	6,932	One 68-in., two 100-in.	72	..	..	100	10,500	18
City of Rome .	Barrow Co. .	1881	543	52 0	38 9	10·432	14·000	22 0	11,230	8,141	Three 46-in., three 86-in.	72	29,286	1398	90	11,900	18·23
Aurania . .	Clydebank Works .	1882	470	57 0	39 0	8·245	12·051	20 9	8,800	7,269	One 68-in., two 91-in.	72	23,284	1001	90	8,500	17·5
Oregon . .	Fairfield Co. .	1883	500	54 0	40 0	9·259	12·500	23 0	10,500	7,375	One 70-in., two 104-in.	72	38,047	1428	110	13,300	18·3
America . .	Clydebank Works .	1884	432	51 0	38 0	8·470	11·520	23 0	9,300	6,500	One 63-in., two 91-in.	66	22,750	882	95	7,354	17·8
Umbria . .	Fairfield Co. .	1884	500	57 0	40 0	8·772	12·500	22 6	10,500	7,718	One 71-in., two 105-in.	72	38,817	1606	110	14,321	20·18
Lahn . .	„ .	1887	448	48 10	36 6	9·174	12·247	23 0	7,700	5,661	Two 32½-in., one 68-in., and two 85-in.	72	..	..	150	8,900	17·78
Paris . . .	Clydebank Works .	1888	528	63 0	41 10	8·373	12·610	23 0	13,000	10,490	Two 45-in., two 71-in., and two 113-in.	60	50,265	1293	150	20,000	21·8
Augusta Victoria	Vulcan Company, Stettin	1889	460	55 6	39 0	8·288	11·795	22 9	9,500	7,661	Two 41⅞-in., two 66⅛-in., and two 106⅞-in.	63	36,000	1120	150	14,110	18·31
Columbia . .	Laird Brothers. .	1889	463	55 6	39 0	8·333	11·860	22 9	9,500	7,578	Two 41-in., two 66-in., and two 101-in.	66	34,916	1226	150	13,680	19·15
Teutonic . .	Harland & Wolff .	1890	565	57 6	42 2	9·826	13·425	22 0	12,000	9,686	Two 43-in., two 68-in., and two 110-in.	60	40,072	1154	180	19,500	21
Normannia .	Fairfield Co. . .	1890	500	57 3	38 0	8·730	13·150	22 0	10,500	8,716	Two 40-in., two 67-in., and two 106-in.	66	46,490	1452	160	16,352	20·78
Spree . . .	Vulcan Company, Stettin	1890	463	51 6	37 6	9·00	12·346	22 0	8,900	6,963	Two 38-in., one 75-in., and two 100-in.	72	..	..	165	13,000	19·6
Fürst Bismarck	Vulcan Company, Stettin	1891	503	57 3	38 0	8·777	13·224	22 6	10,200	8,000	Two 43⅜-in., two 67-in., and two 106¼-in.	63	47,000	1450	157	16,412	20·7
Campania . .	Fairfield Co. . .	1893	600	65 0	41 6	9·231	14·457	23 0	18,000	12,500	Four 37-in., two 79-in., and four 98-in.	60	82,000	2630	165	30,000	22·01
St. Louis . .	Cramp, Philadelphia	1895	536	63 0	42 0	8·50	12·75	26 0	16,000	11,629	Four 28-in., two 55-in., two 77-in., four 77-in.	60	40,320	1144	200	18,000	21·08
Kaiser Wilhelm der Grosse	Vulcan Company, Stettin	1897	625	66 0	43 0	9·46	14·27	28 0	20,880	14,349	Two 52-in., two 89-in., and four 96·4-in.	68·8	84,285	2618	178	30,000	22·5 to 23
Oceanic .	Harland & Wolff .	1899	685	68 5	49 0	10·01	13·98	32 6	28,500	17,274	Two 47·5-in., two 79-in., and four 93-in.	72	74,686	1962	192	27,000	20·72
Deutschland .	Vulcan Company, Stettin	1900	662·9	67 0	44 0	9·89	15·06	29 0	23,620	16,502	Four 36·61-in., two 73·6-in., two 103·9-in., and four 106·3-in.	72·8	85,468	2188	220	36,000	23·25 to 23·5
Kronprinz Wilhelm	Vulcan Company, Stettin	1901	663	66 0	43 0	..	..	29 0	21,300	15,000	Four 34·2-in., two 68·8-in., two 98·4-in., and four 102·3-in.	70·8	93,685	2702	213	36,000	23·25 to 23·5
Kaiser Wilhelm II.	Vulcan Company, Stettin	1903	678	72 0	52 6	9·41	12·9	29 0	26,000	20,000	Four 37·4-in. four 49·2-in., four 74·8-in., and four 112·2-in.	70·86	107,643	3121	225	38,000	23·5
La Provence .	St. Nazaire Works .	1906	597 ft. 1½ in.	64 7½	41 8	9·23	14·35	26 9	19,160	13,750	Two 47·2-in., two 76·2-in.. four 88·18-in.	66·9	58,342	1571	200	30,000	22·05
Lusitania . .	Clydebank Works .	1907	760	88 0	60 4½	8·65	12·56	33 6	38,000	32,500	Turbines	..	158,350	4048	195	68,000	25
Mauretania .	Swan, Hunter, and Wigham-Richardson	1907	790	88 0	60 6	..	..	..	38,000	32,000	Turbines	..	159,000	4060	195	68,000	25

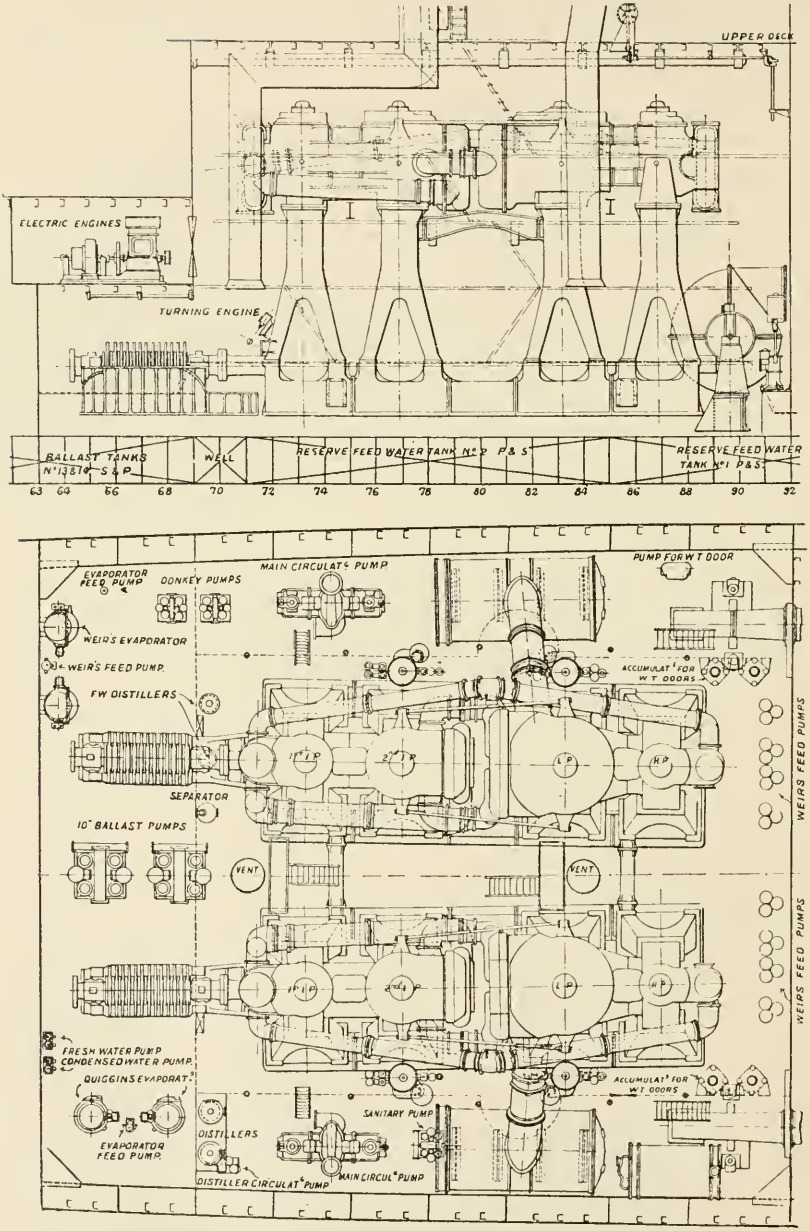


FIG. 148.—Engines of s.s. *Caronia*.

The H.P. and 1st M.P. cylinders are placed tandem, the H.P. being the upper ; and the 2nd M.P. and L.P. cylinders are abaft them in the after engine room, and before them in the forward engine room. These engines are notable for the great power which is transmitted by each shaft, a power which a few years previously was considered too great for any single shaft to carry. The general details of internal construction follow the usual designs, and the proportions are similar.

S.S. "Caronia" (Cunard Line).—Outlines of the engines—four-cylinder, four-stage expansion—of the s.s. *Caronia* are shown in Fig. 148. The *Caronia* is a sister vessel to the *Carmania*, fitted with turbines. Some particulars of interest are :—

Length of connecting rods	12 feet.
Total height from centre of shaft . . .	30 ,,
,, ,, bottom of crank-pits . .	36 ,,
Height of funnel	135 ,,

Howden's system of hot-air draught.

2·35 square feet of boiler heating surface is allowed per I.H.P., and about 17½ lb. of coal are burnt per square foot of grate area per hour, and thus the draught created by the funnel is almost, if not entirely, sufficient, without mechanical draught or other assistance. Coal per I.H.P. per hour (on trial) was 1·29 lb. and the steam consumption 13·4 lb. per I.H.P. per hour, showing that 11·3 lb. of steam were produced for 1 lb. of coal burnt, which gives a very high boiler efficiency of about 82 per cent.

Balancing of Engines.—When an unbalanced engine rotates, there are certain forces transmitted to the bed-plate through the crank-shaft. These forces are, roughly :—1st. The centrifugal forces set up by masses, such as the crank-arms, etc., rotating round the centre line of the crank-shaft and continuously changing direction. 2nd. Forces set up by masses, such as the crossheads, etc., reciprocating in a straight line. If not counteracted or balanced, these forces will cause vibration.

The principles of balancing engines are as follows (Fig. 149) :

1. *Rotating Masses.*—If a mass M rotates with its centre of gravity G at a radius R from the centre of rotation O at an angular velocity of w (radians per second), the centrifugal force will be MRw^2, and this force will be acting along the line OG as shown by the arrow in Fig. A of 149.

This force is obviously constantly changing direction as the mass

revolves, and it is evident that it can be balanced by providing an equal and opposite force, which can be done by causing another mass to rotate with its centre of gravity in the same plane and on a radius formed by producing GO on the side remote from G. The force must be equal to MRw^2, and if m be the balancing mass and r be the distance of its centre of gravity from O, then

$$mrw^2 = MRw^2,$$

in which w will be the same for both because both masses are being rotated by the same shaft.

Therefore $$m \,.\, r = M \,.\, R,$$

from which m and r can be found for a perfect balance.

2. *Reciprocating Masses.*—These are not so simple to deal with. Neglecting the obliquity of the connecting rod, which has comparatively little effect and cannot be taken into account without complicating the calculation considerably, the masses move with simple harmonic motion.

In Fig. B, let O be the centre of rotation, OA the crank, AB the connecting rod and G the centre of gravity of the mass M reciprocating along the line OG. Then, neglecting the obliquity of the connecting rod AB, the mass M may be considered to be at the point C, obtained by dropping a perpendicular from the crank pin centre A on the line of stroke BE, and its motion will be reciprocating between D and E because C follows the motion of the crank.

Now, the acceleration of the point A moving in a circle will be $OA \times w^2$, where w is the angular velocity of the crank arm OA. The component of this acceleration along OB will be $OC \times w^2$.

Therefore, the acceleration of G for the position of the crank shown will be, $OC \times w^2$, and the force causing this acceleration which, if unbalanced, must be counterbalanced by the engine framing, will be

$$M \times OC \times w^2.$$

In other words, the force of inertia of any mass reciprocating in a given line is the component along that line of the centrifugal force of the mass if it be considered to be revolving with its centre of gravity at the centre of the crank pin which is governing its motion. The difference between a reciprocating mass and a rotating mass lies in the fact that a reciprocating mass has no component in the direction at right angles to the line of reciprocation, whereas a rotating mass has a similar component in each direction.

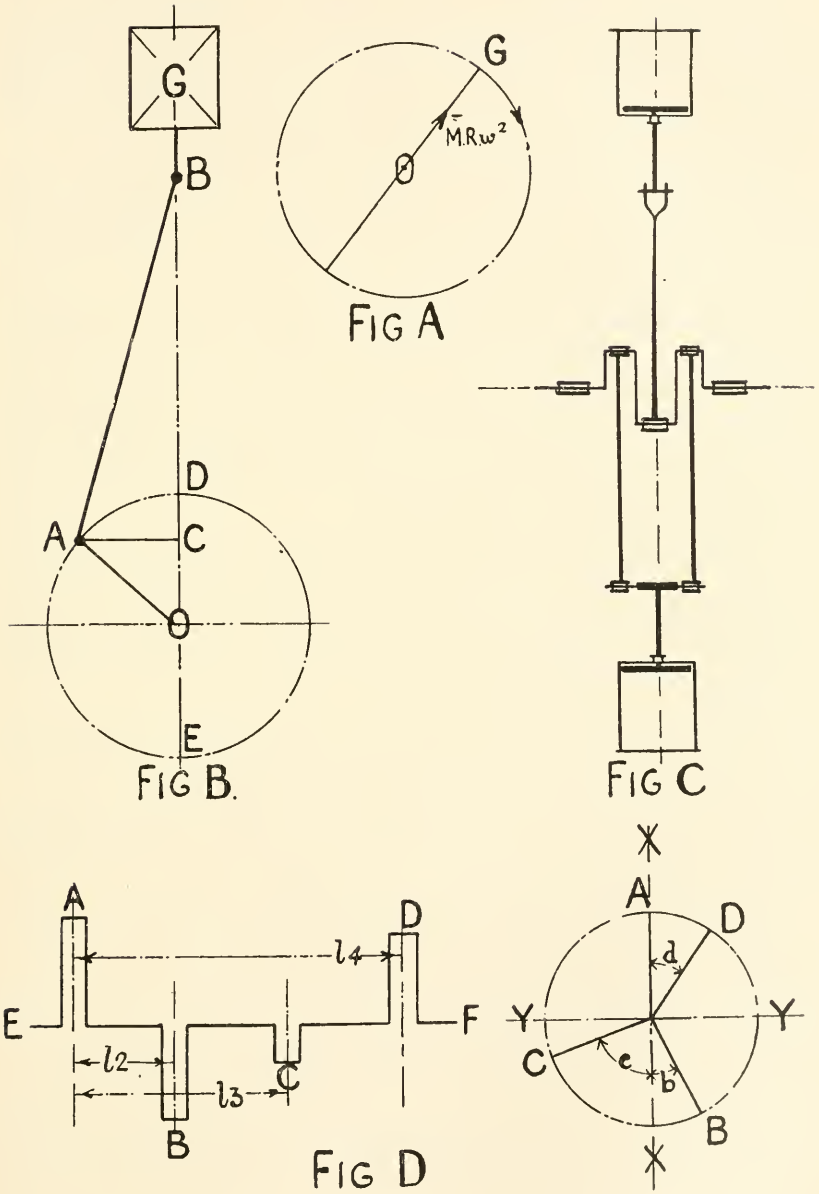


FIG. 149.—Balancing Reciprocating Engines.

It is therefore clear that a reciprocating mass can only be balanced by another mass reciprocating along the same line or by a number of masses reciprocating in the same plane in such a manner that the result of their combination produces equal or opposite inertia forces along the line of reciprocation of the mass to be balanced.

The fact that makes perfect balancing difficult to arrange is that cranks opposed at 180 degrees cannot occupy the same position on a shaft, and when two or more cranks are introduced inertia forces as well as inertia couples come into being, owing to the masses moving at different distances along the crank shaft.

A partial balance for reciprocating masses may be obtained by introducing a revolving mass. If the rotating mass be equal to the reciprocating mass and placed with its centre of gravity at the same radius and directly opposed to the actuating cranks, the forces in the line of reciprocation will be balanced and therefore eliminated, but exactly similar unbalanced forces will be introduced in a direction at right angles to the line of reciprocation.

By reducing the rotating balance weight, a compromise can be effected and vibration will occur in the two directions but reduced in magnitude. This method has been used with success in locomotive engines where vibration in a vertical direction is preferable to that in a horizontal direction.

When considering the balancing of reciprocating masses it must be remembered that when forces are in equilibrium their components in any direction are of necessity in equilibrium, so that masses *reciprocating in the same plane* may be taken as situated at their respective crank pins and balanced as rotating masses and then the components of the centrifugal forces in the lines of reciprocation, which are the forces to be actually dealt with, are in balance.

As rotating masses when once balanced will be in balance when the shaft is rotated to any position, it follows that the same will hold for reciprocating masses, and thus when dealing with balancing it is immaterial in what position the crank shaft is taken to be, therefore for simplicity one crank is usually taken as being on the top centre.

In practice, the introduction of the additional reciprocating weights, or BOB weights, necessary for a correct balance is too complicated to be permissible, so that balancing can only be achieved by balancing one cylinder against another. Thus with a single cylinder engine correct balancing is impracticable.

A two-cylinder engine can only be correctly balanced by placing

the cylinders directly opposed as shown in Figure C, but generally speaking the type to be dealt with is the engine with all its cylinders in the same plane and on the same side of the crank shaft, and this type will only be considered.

A three-cylinder engine can be partially balanced, but with four cylinders a nearly perfect balance may be achieved by suitably arranging the weights of the various parts and the spacing of the cranks along the shaft.

The method of balancing a four-cylinder side-by-side engine is as follows:

In Fig. D, let EF be the crank shaft having cranks in the planes A, B, C, D, and let the cranks B, C, D be placed at distances $l2$, $l3$, $l4$ from A respectively. Consider the crank at A to be at the top of its stroke, and the cranks at B, C, D, to make angles b, c, d, respectively with the line of stroke. Let the rotating masses at A, B, C, D respectively be M1, M2, M3, M4, and the reciprocating masses be m1, m2, m3, m4.

Then, considering reciprocating masses for balance resolved along the plane of reciprocation XX:

$$m1 - m2 \cdot \text{Cos } b - m3 \cdot \text{Cos } c + m4 \cdot \text{Cos } d = 0 \quad . \quad (1)$$

Resolving at right angles to plane of reciprocation, *i.e.* along YY:

$$m2 \cdot \text{Sin } b - m3 \cdot \text{Sin } c + m4 \cdot \text{Sin } d = 0 \quad . \quad . \quad (2$$

Taking moments about A, vertically:

$$-m2 \cdot \text{Cos } b \cdot l2 - m3 \cdot \text{Cos } c \cdot l3 + m4 \cdot \text{Cos } d \cdot l4 = 0 \quad . \quad (3)$$

Taking moments about A, horizontally:

$$m2 \cdot \text{Sin } b \cdot l2 - m3 \cdot \text{Sin } c \cdot l3 + m4 \cdot \text{Sin } d \cdot l4 = 0 \quad . \quad (4)$$

And, considering similarly the rotating masses:

$$M1 - M2 \cdot \text{Cos } b - M3 \cdot \text{Cos } c + M4 \cdot \text{Cos } d = 0 \quad . \quad (5)$$

$$M2 \cdot \text{Sin } b - M3 \cdot \text{Sin } c + M4 \cdot \text{Sin } d = 0 \quad . \quad (6)$$

$$-M2 \cdot \text{Cos } b \cdot l2 - M3 \cdot \text{Cos } c \cdot l3 + M4 \cdot \text{Cos } d \cdot l4 = 0 \quad . \quad (7)$$

$$M2 \cdot \text{Sin } b \cdot l2 - M3 \cdot \text{Sin } c \cdot l3 + M4 \cdot \text{Sin } d \cdot l4 = 0 \quad . \quad (8)$$

and any solution which satisfies the above equations will give a perfect balance.

The problem may also be solved graphically, using the polygons of forces and moments.

When balancing an actual engine the slide valves and gear must be allowed for, and may best be dealt with by considering the eccentrics

as cranks and dividing the actual weights by the ratio of the crank radius to the throw of the eccentric so as to bring the forces involved to the same proportion as for the cranks. The slide valves and gear are not allowed for in the above calculation for the sake of simplicity.

The connecting rods are taken as partly rotating and partly reciprocating in the proportion obtained by finding the position of the centre of gravity of the rod, which is easily determined by slinging the rod. If x be the distance of the centre of gravity of the rod from the centre of the crosshead pin, and the length of the rod from centre of crosshead to centre of crank pin be l, then xl of the weight is considered as rotating and $\frac{l-x}{l}$ is considered as reciprocating.

Weights of Machinery and Boilers.—For purposes of comparison, the years 1904, or ten years before the war; 1914, immediately preceding the war; and 1919, immediately following the war, are taken. 1904 may be taken as the beginning of the Dreadnought period and the general introduction of turbine machinery, and the lighter type of water-tube boilers, for naval purposes, but many naval vessels had already been fitted principally as an experiment before this time. The reciprocating machinery of the six *Duncan* battleships was designed for 18,000 I.H.P. and with 24 Belleville boilers weighed about 1600 tons, giving an output of 11·25 I.H.P. per ton of total weight. The two *Triumph* class, with a stroke of 39″, instead of 48″ in the *Duncan*, and with 25 Yarrow boilers, gave a designed output of 12·5 I.H.P. per ton of total weight.

In 1914 the turbine machinery of battleships developed about 30,000 S.H.P. and a designed output of about 13 S.H.P. per ton of total weight was obtained. During the war the S.H.P. increased to about 40,000 with an output of about 20 S.H.P. per ton, or nearly double that of fifteen years earlier.

In 1904 the armoured cruisers *Minotaur* and *Shannon*, of 27,000 I.H.P., obtained an output of 11·1 and 11·2 respectively I.H.P. per ton of total weight, and only a saving of 20 tons was effected by fitting 24 Yarrow boilers in the *Shannon* in place of 25 Babcock boilers in the *Minotaur*; to-day the difference would be far greater as only 3 or 4 Yarrow boilers would be required for the S.H.P. of the *Shannon*.

In 1914 battle cruisers had taken the place of armoured cruisers and the S.H.P. had increased to 75,000 with an output of 19 S.H.P. per ton. During the war the S.H.P. increased to 110,000 and the output was 30 S.H.P. per ton. Some further development to possibly

180,000 has taken place ; figures for comparison of total weight and output are not available, but the probable output is of the order of 35 S.H.P. per ton of total weight.

In 1904 light cruisers were principally represented by the *Pathfinder* class with reciprocating engines and upright water-tube boilers which gave an output of about 20 I.H.P. per ton of total weight. Ten years pre-war development of turbines, beginning with the *Amethyst*, enabled an increase of power to 25,000 S.H.P. to be obtained with an output of about 37 S.H.P. per ton of total weight. During the war the S.H.P. increased to 40,000, with an output of about 45 S.H.P. per ton ; the general increase in speed was about 10 knots, and in some few instances the S.H.P. increased to 60,000 with a speed of over 37 knots.

The destroyer, since its first introduction in 1894, has contributed as a forerunner in reducing the weight of machinery and boilers in all types of vessel. In 1904 many destroyers with reciprocating engines of short stroke had an output of about 45 I.H.P. per ton of total weight of machinery and boilers. During the next ten years turbines took the place of reciprocating engines and the power was increased about four times. With 25,000 S.H.P. an output of about 64 S.H.P. per ton was obtained in 1914. Nearly 300 destroyers and flotilla leaders were built during the war, and the S.H.P. was increased to over 30,000 in destroyers and 43,000 in leaders, with a corresponding output of about 75 S.H.P. per ton.

During the ten years 1904 to 1914 these increased outputs arose principally from the adoption of direct-driven turbines and water-tube boilers of a more efficient type. During the war period the increase arose principally from the adoption of the geared turbine and the more extended use of oil fuel and superheated steam. Other causes contributed materially in the same direction, including condenser apparatus, but these are more in the nature of continuous development rather than fundamental alterations in the class of machinery and boilers. The various comparative improvements are stated elsewhere, and those effected by turbines and gearing in Chapter XXII.

With regard to *Mercantile machinery* it is difficult to arrive at any definite conclusions on the rate of progress in output of power per ton of total weight of machinery and boilers. Possibly for reciprocating machinery and cylindrical boilers the increase may be estimated as about 7 I.H.P. in 1904 to 10 S.H.P. in 1919 per ton.

During the war many mercantile vessels were fitted with standard

engines, both of the reciprocating and turbine types with gearing, but no accurate figures of comparison are available. A rough estimate would appear to show that with double reduction gearing and cylindrical boilers an output of about 20 S.H.P. per ton might be comfortably obtained, and if water-tube boilers were substituted for cylindrical boilers the output might be expected to reach at least 25 S.H.P. per ton. Figures given by engineering firms, although undoubtedly accurate in themselves, are not always a fair basis of comparison because it is not known what weights are included, whereas in a naval comparison the weights include practically everything fitted into the main machinery and boiler spaces.

Many shipbuilding and engineering firms are ready to put their war experience of fitting both geared turbines and water-tube boilers at the disposal of shipowners, and future progress therefore rests principally with the latter. Under the white ensign many mercantile engineers have gained experience and a favourable insight into the working of turbines and the treatment of water-tube boilers and their contingent auxiliaries, and therefore the road lies much clearer and safer than before the war when only a very select few had been inculcated with the necessary experience. On the adoption of oil fuel the decision is less easy because supplies may not be available in the ports of call and there is the very serious consideration of price of fuel. Roughly stated it does not pay to burn oil when its price per ton exceeds two and a half times that of coal.

For cargo vessels of moderate speed, 11 to 12½ knots, and power 2500 to 4000 B.H.P., the most economical method of propulsion is by means of the Reciprocating Internal Combustion Engine (Chapters XXXVII. and XXXVIII.). Makers claim also considerable saving in weight and cargo space, as well as engineering staff.

CHAPTER XVII

CYLINDERS AND THEIR FITTINGS

Cylinders.—The cylinder is a very complicated casting, and it is almost impossible to make it satisfactorily in steel, so that cast iron is generally employed, stayed as necessary to give it local strength.

When the piston is descending, the cylinder has to withstand the resistance of the upward pressure opposed to the piston ; this means that the cylinder must be supported in such a way that it cannot yield to this pressure, and must be securely held down. In a similar way it must be capable of withstanding the downward pressure when the piston is rising ; consequently the method of securing it down to the supports on which it rests must be proportioned to these stresses. In Fig. 150 the cylinder feet are clearly shown ; the rear foot is generally secured to the back frame, or column, which also supports the guide, and the front supports are generally cylindrical columns or pillars.

The bottom or front end of the cylinder generally forms part of the casting, but covers are fitted at the back end so that the piston and slide valve, and, if required, the liners, working barrels, and false faces, can be admitted.

The H.P. cylinder and slide casing are generally made in one piece of hard close-grained cast iron, with feet, as shown in Fig. 150.

The M.P. cylinder and slide casing follow the same course, where a cylindrical valve is employed, as is now usual ; but if a flat valve is fitted, then the casting follows the practice for the L.P. cylinder and casing, which is made as shown in Fig. 151.

Cylinder False Face.—The working face of the cylinder is lined where the slide valve moves on it, in a manner somewhat similar to that provided for the piston.

When cylindrical valves are employed, as is now usual for the H.P. and M.P. slide valves, the face is made in two parts, as shown in Fig. 150. The inner, or bottom, face is a cylindrical barrel, fitted

with the necessary steam ports and an inward flange, which is jointed to the cylinder casting in the same way as the lower end of the cylinder liner. This inner face must be of slightly smaller diameter than the outer one, so that it may be passed into its place. The outer, or upper, face is made with an outward flange at its top end ; this is necessary for getting the valve into place, and also for its subsequent working. The joints are each made with red-lead in the usual way, and the bolts

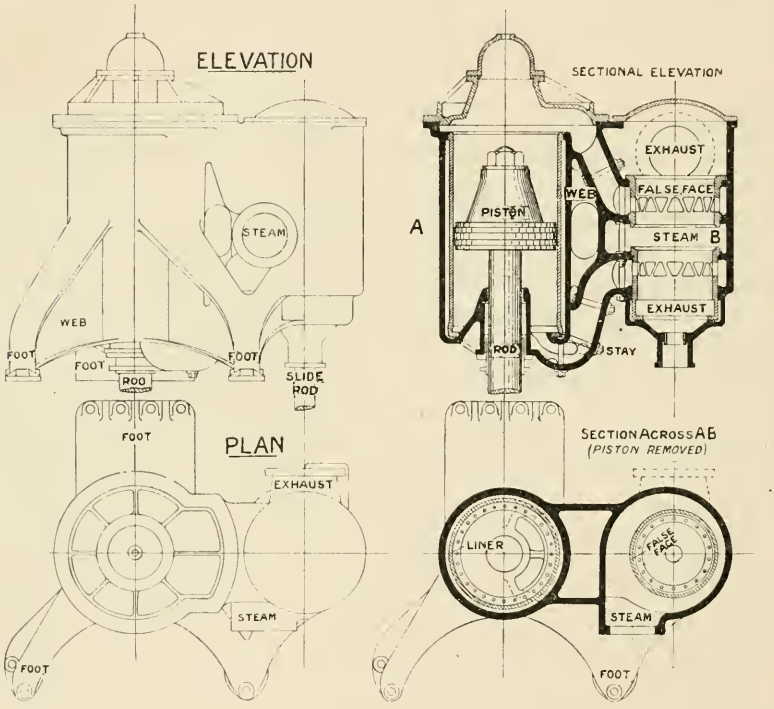


Fig. 150.—H.P. Cylinder.

are secured by either guard rings and studs, or the bolt heads are caulked at one corner. Separate parts are preferred for the upper and lower faces, because, if made in one piece, there is some difficulty from unequal expansion in making both joints steam-tight. The H.P. and M.P. false faces are made of hard close-grained cast iron.

For flat valves, the face is made in one piece (Fig. 151), and secured to the cylinder casting by cheese-headed screws. The same material is used for the other false faces.

Steel Stays, as shown in Fig. 150, are used for the flat surfaces subject to pressure in the cylinder ports (or steam ports, as they are more usually called). These stays are fitted with nuts on the outside, and bosses are made on the casting for bringing up the strength locally —to at least an equivalent to the stays employed. In some cases the stay is fitted into a long-sleeved hole, so that it is not in actual contact with the steam. The *steel* stay and the *cast-iron* sleeve expand unequally, and stresses are thus brought on the casting of the cylinder ;

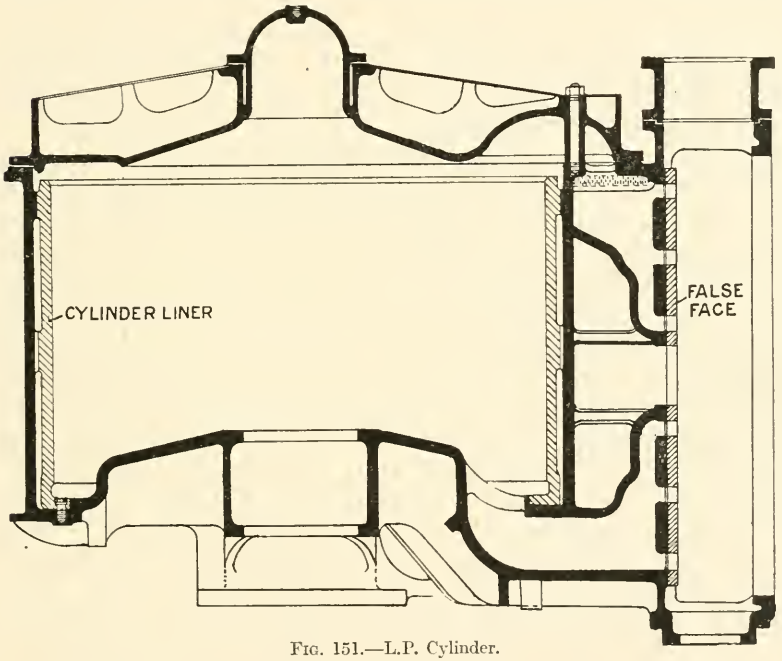


FIG. 151.—L.P. Cylinder.

therefore the naked stay is preferable, although a steam joint has to be made at its outer end. The nuts are generally riveted over on the end of the stay. to prevent their slacking back.

In addition to the stays, *webs* are made wherever possible in the casting, to give local strength to it ; but care must be taken that this webbing and staying are consistent with the expansion of the cylinder itself, and that they shall not induce fracture by their unequal expansion when the engine is warmed up to its work ; many fractures have occurred from this cause alone.

The cylinder **covers** are made of cast steel. In a few cases, however,

they have been made of forged or stamped steel; but this is only possible when the form is simple, and without any complications of webbing. The interior of the cover follows the shape of the piston as closely as possible; this reduces the clearance spaces. The exteriors are made with webs of T section for those fitted radially from the manhole flange (Figs. 150 and 151). A vertical web, sometimes of inverted L section, is made far enough from the edge to leave the heads of the nuts clear of it; this follows the shape of the edge of the cover.

Lagging.—All projections on the cylinder casting are made so that it is not necessary to disturb the lagging when fitting or jointing up the connections, steam or eduction pipes, drains, ramming stays, indicator cocks and pipes, jacket valves and drains, etc.

The whole of the hot surfaces of the cylinders, slide-valve casings and covers, cylinder covers and pipes, are covered with asbestos or other bad conductor of heat. Mats of asbestos fibre are used wherever possible, and all parts are then covered over with asbestos cloth. To keep the lagging in place it is covered up with strips of closely fitting mahogany (or sheet lead for bottoms of cylinders), bound together with brass ribbons and screws. The foundations to which the outer strips are fixed are preferably made of some incombustible material—not wood—and a form of asbestos is generally used.

Cylinder Liners.—The working surfaces on which the piston and slide valve move are generally made separate from the cylinder casting; this reduces its complexity, and provides more suitable material for the working surfaces.

The L.P. cylinder liner, or working barrel, is made of hard close-grained cast iron (Figs. 151 and 153).

The H.P. and M.P. cylinder liners are made of forged steel; and are thus made thinner for the greater pressure which they have to stand, than if made of cast iron (Fig. 150).

The space between the cylinder and the liner is utilised as a *jacket steam-space*, generally from 1 to $1\frac{1}{4}$ inches in width. To prevent the steam in the jacket leaking into the cylinder, and *vice versa*, it is necessary to make the ends of the cylinder liner steam-tight at the inner surface of the cylinder.

The inner end (the bottom in a vertical inverted engine) of the liner is flanged inwards, and fits accurately into a recess cut for it at the end of the cylinder, and makes a joint with it, as shown in Figs. 152 to 153. The type shown in Fig. 152 is now usually employed for

all cylinders ; the tops of the heads of the bolts are below the level of the flange. The jointing material is a thin red-lead mixture, with a strand or two of asbestos twine to prevent it from being blown out of its place ; but where the face of the flange or cylinder is not very true, it is better to use asbestos sheeting, about $\frac{1}{16}$-inch thick. The bolts are made with square heads, and are tightened up by using a long spanner from outside the cylinder ; but, if possible, a short socket spanner and tommy are more effective. To prevent the bolts slacking back, they are caulked in at a point on the collar ; and if nuts are fitted on the outside, split pins are fitted, to prevent the nuts slacking.

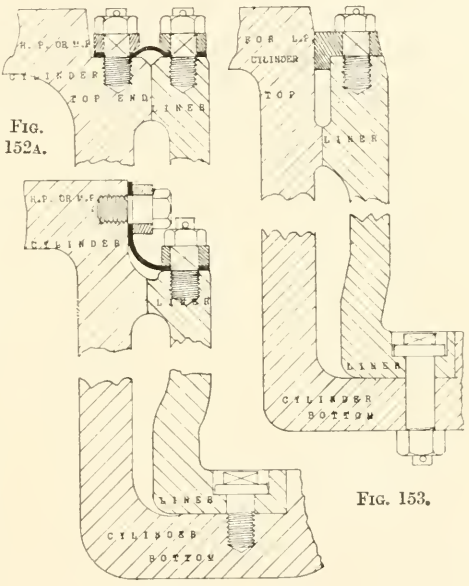


Fig. 152A.

Fig. 153.

Fig. 152.

The outer end (or top, in a vertical inverted engine) of the liner cannot be jointed in the same way as the bottom end, because there is unequal expansion of the steel of the liner and of the cast iron of the cylinder to be allowed for, and therefore an expansion joint is fitted at this top end.

Various methods are shown in Figs. 152 and 153. In addition to making a tight joint, allowing for independent expansion of the cylinder and liner, it is preferable to make it capable of adjustment and repair without disturbing the flange joint of the liner.

For the H.P. and M.P. cylinders a flat ring, either with or without a single upward (Fig. 152) or downward corrugation, is the best for ordinary use, and most easily jointed up ; but it entails a slightly increased diameter of cylinder, at the flange, to contain it. The copper expansion ring is stiffened under the nuts by two iron or steel annular washer rings, as shown, and these are rounded off on one edge to allow movement of the expansion ring. The studs securing the rings are made with square, or pear-shaped, necks, where they pass through

the supporting rings, to prevent them slacking back. In some cases, collars are made on the studs under the ring; but if the previous method is used, the collars are unnecessary, although frequently fitted. The copper angle ring, with a well-rounded corner (Fig. 152), is used where space and weight are of more importance than convenience in making the joints. The bolts and studs are in this case fitted alternately in the joint on the cylinder and the joint on top of the liner. The liner joint must be made first, and the difficulty of making and spreading the two joints so that they shall both be efficient is manifest. The studs are secured in the usual way, and the corners of the heads of the bolts are caulked, to prevent them slacking back.

For the L.P. cylinder a ring with a flange, as shown in Fig. 153, is used to press down the packing caulked into the recess below. The packing used is generally asbestos rope, well covered with black-lead, and smeared with sufficient mineral grease to make the black-lead adhere to the packing. This is the simplest form of *liner expansion joint*, and is generally sufficient for low pressures; but it is not thought good enough for the H.P. and M.P. cylinders, although experience has proved it reliable.

Pistons.—A cast-iron piston of the flat or hollow type, which was frequently fitted, is now seldom used in large cylinders. For small engines it is still very common, either with or without a junk ring, the use of which is explained later. The front and back faces of a flat piston are strengthened by radial webs between them, and for moderately high pressures and diameters one face, generally back, is coned to give increased strength.

As pressures increased, it became impossible to build flat cast-iron pistons of sufficient strength without a great increase of weight, so that now all pistons for large engines are made of steel, and generally forged into the shape of a cone, which is unyielding in shape under the pressure to which it is subject. This also embodies a minimum weight. The height of the cone necessary for the largest piston is first determined, and then this height is generally adopted for the remainder, so that all the rods shall be of the same length, and also interchangeable if of the same diameter. This height is generally sufficient for the necessary strength of all the pistons of the same engine.

Types of pistons are shown in figures: Fig. 150 for a modern H.P. cylinder, and in Fig. 154 for the L.P. cylinder of a large modern engine. For the M.P. cylinder, the same type as Fig. 150 is generally fitted.

Attachment of Piston to Rod.—The method now generally adopted is shown in Fig. 154, and is the same for all cylinders. The piston is accurately bored to a cone of one in four, and the remainder of the upper part *SN* of the hole is made parallel. The cones are now standardised for each size of rod. The rod is made to fit this hole accurately on the cone; and to facilitate the fitting at the top and bottom of the cone, a part, *M*, at its middle is sometimes cut down to a rather smaller diameter. The cone is carefully scraped away

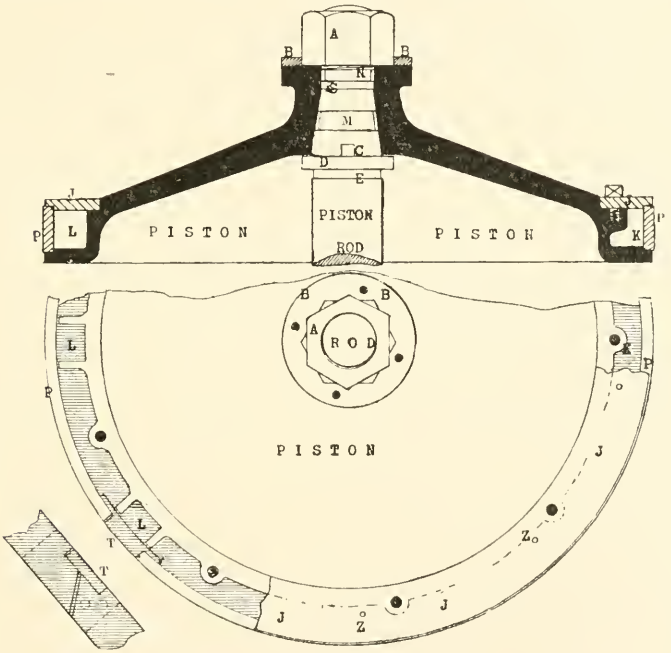


FIG. 154.—Piston-rod End and Piston.

until it allows the rod to enter; when the nut on top is hardened up, the flanged collar *D* is just clear of the piston about 0·015 inch. The object of this fitting is to make certain of the piston being a proper fit on the rod; and that, if by any undue strain it be driven further on to the cone, the collar comes into play, and prevents the piston being split or fractured.

The nut *A* is made of wrought iron, so that it shall not seize on the steel rod. It is prevented from slacking back by the arrangement shown, where *B* is a thin plate, with its corners touching the nut, and

held in place by studs fitted with square or pear-shaped necks, and nuts with split pins. Another plan is to use a cotter-shaped key through the end of the rod above the nut, and partially sunk into it; the rod can be a little shorter with the previous arrangement.

Under the collar *D* a clearance space, *E*, is sometimes cut, so that the metal of the rod cannot be worked up to a shoulder. In some cases the presence of a collar on the rod necessitates the splitting of the neck ring and packing glands, so as to get them in place. With double gudgeon pins this is unnecessary.

A stretching length is made at *N*, where the rod is turned down to a diameter just less than the bottom of the thread of the screw. When the piston is ascending, there is a great local stress on the cross-sectional area at the root of the screw thread at the junction of the piston and nut; any tendency to stretch is naturally taken at this part, and acts on a very short length. The stretching length, which is of slightly less area, tends to spread this tension over a greater length, and thus increases the actual strength on slightly less weight.

Stretching lengths are also made in the bolts used for the connecting rod and main bearings, for similar reasons.

A stop pin, *C*, is fitted to prevent the piston turning on the rod. Recesses are made in the piston for the necessary fittings—packing rings, junk ring, springs, and tongue piece; and, if possible, it is arranged for the piston to be self-draining, so that no water can lodge on it.

For *removing the piston from the rod* and taking its weight, various methods are employed, but they all embody the same principle. In Fig. 155 a screw thread is cut on the top end of the piston; after the nut is removed, a cap, *A*, is screwed on this, and an eyebolt, *B*, is screwed through it, with its point bearing on the end of the rod *R*; by screwing the eyebolt further in, the piston rod is forced back and freed from the piston *D*. For the piston shown in Fig. 154, a flange is made at the top of the piston; under this flange two half-clips are bolted together, and then two long bolts with a strongback are used to support a central screw. For small pistons two or more holes are drilled in the upper end of the piston, covered by the nut when in place; studs fit into these holes, and with a strongback a central screw can be used as in the previous cases. The forcing bolt, or eyebolt, is generally used for lifting the piston after it is removed from the rod.

In some cases the rod has been withdrawn from the piston by fitting packing, generally wooden chocks, between the piston and the

cylinder end, and then pulling the rod away by turning the engine by hand. This method should not be used, as it brings unknown strains on the cylinder, and may be dangerous.

Piston Packing and Fittings.—The piston, in addition to being made strong and stiff enough to resist deformation under maximum load with minimum weight of material, and being strongly attached to the piston rod, must also be made to work steam-tight in the cylinder with minimum friction. This is arranged by fitting the body of the piston with metallic packing, so that the body itself never comes into actual contact with the metal of the cylinder liner.

The *junk ring* is used to keep the packing in place (*J* in Figs. 154

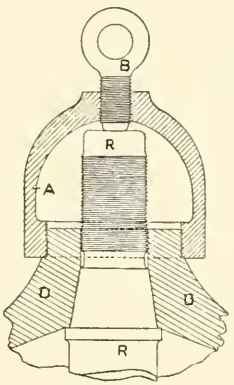


FIG. 155.—Removal of Piston from Rod.

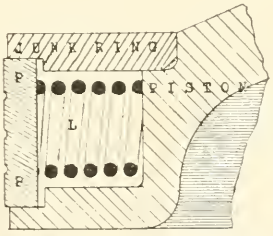


FIG. 156.—L.P. Piston Packing.

and 157), and it is secured to the piston by studs and nuts as shown; a *guard ring*, *G*, is fitted to prevent the nuts slacking back. The studs, *Z*, in both the junk and guard rings are fitted with square necks; and split pins are fitted for the guard-ring nuts, to prevent them slacking back. The advantage of fitting a junk ring is that the packing can be examined and fitted without removing the piston from the cylinder. The junk ring should have about 0·005 clearance and should not press hard on the packing, and it should be possible to move the packing ring round comfortably by hand after the junk ring is tightened up in its place. The packing ring should, however, be sufficiently tight to keep the steam from leaking past it, and at the same time it must allow the ring freedom to move up to its working face on the liner; the sides of the ring are therefore frequently scraped up to a true surface where they butt on the flange of the piston and on the face of

the junk ring. The junk ring is made of forged steel (Fig. 156); but if, as in Fig. 157, it is allowed to come in contact with the metal of the cylinder liner, it may then be made of cast iron.

The most common form of **piston packing** is shown in Figs. 154 and 156, in which it consists of a single broad ring of cast iron or phosphor bronze, *P*, forced out by a gentle and uniform pressure by springs, *L*, behind it. The springs are now of the spiral type, but coach springs were used in the earlier engines. The packing ring projects radially about $\frac{1}{8}$-inch beyond the body of the piston and of the junk ring. The spiral springs are fitted into recesses, *L*, behind the ring, and the pressure exerted by them on the working face should be about 2 lb. per square inch of ring surface for the L.P. cylinder, and, if used for the M.P. cylinder, about $2\frac{1}{2}$ lb. per square inch is allowed. The springs are about $2\frac{1}{2}$ to 3 inches in diameter, made of wire $\frac{1}{4}$ to $\frac{3}{8}$ inch in diameter; and one spring is fitted at a point midway between each pair of junk-ring bolts, or about 9 inches apart.

The packing ring is cut diagonally (*T* in Fig. 154), and a *tongue piece* is fitted to close the gap, and at the same time allow for expansion of the ring and for wear. The tongue piece is generally fastened to one end of the cut ring by two or more countersunk screws; the other end is left free to move inside the ring and along its edge, as shown.

For the H.P. and M.P. cylinders, as a rule, no springs are used to press out the packing against the liner. The packing rings are made a little larger in diameter than the liner which they are eventually to fit; they are then cut, generally diagonally, and fitted with tongue pieces if required. Sufficient length is cut away from each ring to allow it to enter the cylinder liner, and at the same time leave sufficient space for expansion between the ends of the ring at the cut. The ring itself tends to keep its natural shape, and in consequence springs out against the liner, and is generally steam-tight against it.

Ramsbottom rings fitted to a piston are cut and made in the way mentioned above, but without tongue pieces. The divisions are spaced around the piston so that no two are opposite, and with very fast-moving engines the leakage is inappreciable. These rings are generally made of hard phosphor bronze.

The more recent types of H.P. and M.P. pistons are fitted with **restrained rings** of hard bronze (Figs. 157 and 158). The tendency of the packing rings, *P*, to spring outwards is restricted to a small amount by a *tension ring*, *R*, made of cast iron. *R*, is solid and uncut, with a very slight clearance, $\frac{1}{64}$ to $\frac{1}{32}$ inch, between it and the body of

the piston, and about $\frac{1}{32}$ to $\frac{1}{16}$ inch between it and the packing rings, diametrically. The flange of the piston is made so much smaller in diameter that the tension ring prevents it coming in contact with the

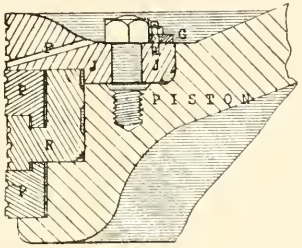


FIG. 157.

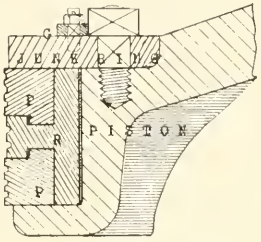


FIG. 158.

Restricted Packing Rings.

cylinder liner. The piston in Fig. 157 is fitted with drain holes, *F*, as it is not naturally self-draining.

Fig. 159 shows Lockwood and Carlisle's double-action metallic packing rings and springs. The packing rings, *PP*, are strongly pressed away from each other by the continuous spring *U*, to make a steam-tight joint at the faces of the junk ring and the flange of the piston. Each ring is fitted with a tongue piece, which completes the continuity of contact. The spring is locked, and the rings overlap its edges, which holds them in position and restrains the steam from

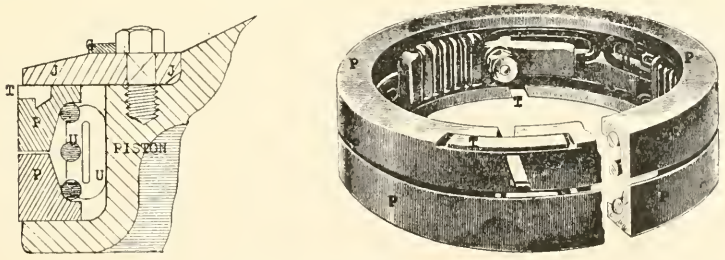


FIG. 159.—Carlisle Piston Packing.

unduly pressing the rings against the working barrel of the cylinder. When required, the spring can be enlarged in diameter by removing the screw *N*, and lengthening the hole in the plate to allow for the requisite liner, after which the screw is replaced. Very satisfactory results are obtained with this packing.

This type is also used for cylindrical slide valves, with certain modifications to make it suitable.

Uncut or solid rings are fitted to some small pistons. About $\frac{1}{200}$-inch is allowed between the ring and the cylinder, which is generally made without a liner in small engines; the clearance allowed makes the ring a fairly good working fit when the engine is warmed up to its work. In all cases the rings should be tried separately in the cylinder or working barrel from top to bottom of the stroke, to see that they are a proper working fit, and at the same time can move freely, when worn, up to the working face; this is particularly necessary with uncut rings. The working barrel is generally bevelled off at the top and bottom of the stroke, so that the ring partially overruns the parallel part at each end; this prevents a shoulder being worked up at the ends of the stroke.

For small pistons, such as dynamo engines, etc., good emergency results were obtained by the author with solid piston packing made of white metal composed of 7 per cent each of copper, tin and lead, and 79 per cent of zinc. The cylinders were warmed and the piston then made a good fit just movable by hand pressure. No appreciable wear was observed over long periods of working.

Water grooves are now fitted to nearly all packing rings, whether for steam pistons or for pump plungers. These are shown in some of the figures. When a junk ring or tension ring is fitted in such a way that it may come in contact with the working surface of the barrel, water grooves are cut in it also. The grooves should be spaced about $\frac{3}{4}$ to 1 inch apart, and cut from $\frac{1}{8}$ to $\frac{1}{4}$ inch wide and deep, rounded semicircularly at the bottom of the groove. Their object is to form a space where a film of water can be held to assist the lubrication of the packing ring; their usefulness is now well established from some years of practice.

Piston-rod Packing.—Where the piston rod passes through the end of the cylinder, it is necessary that it should be steam-tight, and a packing gland is fitted at this part (Figs. 150 and 206).

Next to the cylinder interior a **neck ring** is fitted, which is a moderately good fit on the rod, and made of gun metal. This forms the bottom of the gland—actually the top in an inverted engine; it is perhaps better to term it the *inner* end. The neck ring is clearly shown in Fig. 160, which is an enlarged view in section of a packing gland and a type of packing made by the *Combination Metallic Packing Coy.*

The packing consists of gun-metal coned and telescopic holding rings, *BB*, which form the seating for the working rings *CC*, made of

white metal, and coned in the opposite way. A distance piece, *D*, is inserted to keep the whole of the rings in place, and this distance piece forms the seating for another packing. The metallic packing is pressed on to the rod by spiral springs, *M*, of special shape, suitable for their position, in a spring holder, *A*, next the neck ring. The neck ring in any engine cannot be made a very tight fit, so that a little steam leaks past it and presses on the inner ring, which in its turn keeps the white metal rings up against the rod, and so relieves the springs of their work after steam once finds its way into the recess between the neck ring and the inner packing ring. The pressure on the rod exerted by the packing is therefore proportional to the pressure in the cylinder, and this may reduce the friction to a minimum. The whole of this packing is contained in hollow metal boxes, and this arrangement permits a better adjustment and fitting of the packing, because it can be inserted complete into the stuffing box.

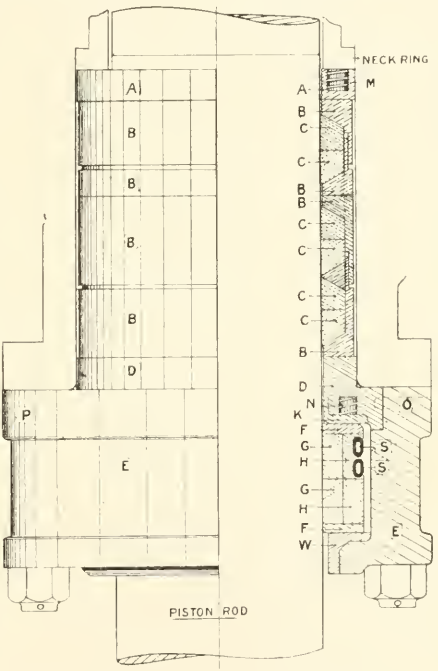


FIG. 160.—C.M. Packing, "H.H." Type.

Sufficient clearance is allowed between each pair of rings for wear and consequent tightening up of the glands. The outer or containing rings, which are made in two halves, are held together by screws in some cases, when it is not necessary for them to touch the stuffing box on the outer rim. This allows a little side play in the rod, and is preferable. If the packing is well fitted and the rod runs truly in line, there is little trouble in keeping a steam-tight joint, and good results are obtained. The white-metal rings are made in three or four parts, and care must be taken that the ends of contiguous rings are not in line, and that they cannot butt when warmed up and the engine is working. No adjustment of the metallic packing can be made when the engines

are at work. Water grooves about $\frac{1}{2}$-inch apart are usually cut in the white-metal rings.

The outer packing consists of two rings, *G*, of wearing metal contained within two other rings, *H*, which, by means of springs, *S*, fitted on the outer circumference, press the wearing rings *G* on to the rod. Vertical movement of the rings *G* and *H* is prevented by two face rings, *F*, one above and below, and above the upper face ring is another ring, *K*, pressed downwards on to the packing by spiral springs, *N*. Below the lower ring *F* is a distance angle ring, *W*. The outer packing is held in place by a box, *E*, which is secured to the upper packing box by studs and nuts. Oil can be introduced through a hole, *O*, to the rod and packing, and opposite it a drain, *P*, may be fitted for carrying away any excessive water or pressure—when the engines are standing under steam, for instance.

Mixed Packing for Rods and Spindles.—In many cases, particularly for auxiliary engines, the ordinary metallic packing is not altogether suitable, and soft packings are used. These, although useful in preventing leakage, are not conducive to a good appearance of the spindles, nor are they very efficient from a point of view of frictional resistance. Recently a fairly good packing of the graphite type has been introduced to the naval service, and it requires some discretion in its use. For a rod moving truly in line and in clean condition it is advisable to fit a closely fitting neck ring either of white metal or gun metal, and then the graphite packing is not forced into the cylinder. In some cases the gland may require also to be closely fitted to prevent the packing being forced into the space between it and the rod. If the rod cannot be brought into line and turned up, it is advisable to fit elastic core packing for the inner and outer turns, with the graphite between them.

For glands of expansion joints, of pipes, etc., graphite may be used in the same manner ; and failing graphite, alternate turns of elastic core and asbestos, the inner turn being elastic core, makes a good and durable joint.

For large rods, piston rods, and others up to 9 inches in diameter, subject to a vacuum, the author designed and fitted packing of the mixed white metal and soft types, as shown in Fig. 161. This allows for some lateral play of the rod, and by fitting a false neck ring (in halves for very large rods) with a shallow cone, the heavy pressure is relieved from the rod. It appeared to be entirely satisfactory for L.P. piston-rod packing, and a decided improvement in vacuum and appearance of the rod was obtained with it in comparison with soft packing

alone ; on the whole, there was some reason to think it superior to a complete set of metallic packing, and it is cheaper. In all cases asbestos packing should be withdrawn from glands if the engine is not to be under steam again within a few days, otherwise the rods are marked and finally become scored.

For the adjustment of soft packing it is arranged that the nuts *A*

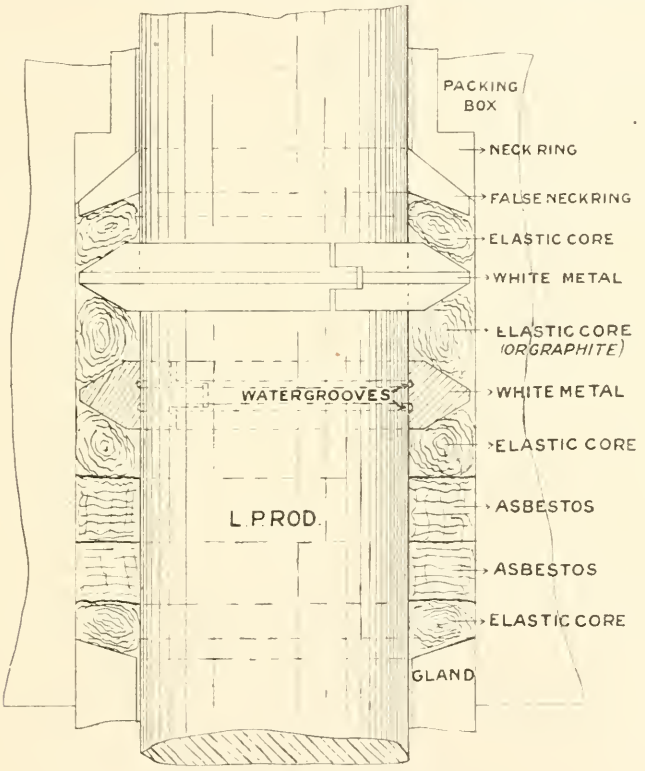


FIG. 161.—Mixed Metallic and Soft Packing.

(Fig. 162) on the studs can be all tightened up together by means of cogs fitted to each nut, and the whole of which are in gear with a central cogged ring, *B*, fitted to revolve about the gland *C* as an axis. This gearing, *G*, is always made workable from the starting platform. The flange *D* of the gland on which the cogged ring is fitted is cupped on its upper side, into which oil can be run for lubricating the piston rod. Just below is fitted a third gland, *E*, packed with a soft gasket packing,

which is used to retain the oil on the rod and to prevent it running away; some, of course, works through this gland, but the greater part is retained.

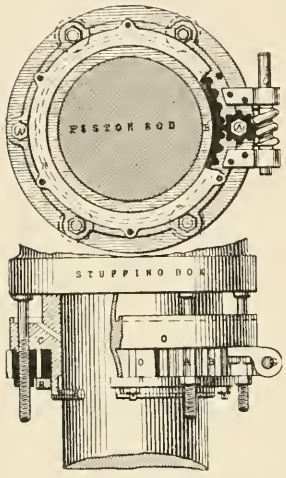


Fig. 162.
Gland tightening-up Gear.

United States Duplex Packing.—For over 150 lb. gauge-pressure a double packing is more efficient, and the type shown in Fig. 163 is frequently fitted. The packing nearest the piston is of the white-metal coned type shown in Fig. 160 and tends to wire-draw the steam, while the block packing below it ensures steam-tightness.

A cupped plate, *A*, supports springs *B*, which press on the flange of a plate *C*. (In some cases *C* is prolonged upwards, and then forms the neck ring.) *C* presses on a coned gun-metal ring *D*, which compresses the coned white-metal rings *EE* (each cut in three parts) on the rod. The (rectangular section) white-metal ring *F* tends to block the passage of steam at the nearly butting ends of the parts of the rings *EE*. The rings *D*, *E*, *F* are contained in a gun-metal casing *G*, pressed down on *K*, which makes a ball joint with the socket piece. A clearance of about $\frac{1}{32}$-inch between *A* and *C* allows a slight leakage into the space about *G*. A drain is fitted to this space for occasional use.

The block packing, which is alone necessary for moderate pressures, is compressed between the two plates *MM* by vertical springs fitted in the socket plate, and pressing on the plate L, which has a projecting lip. This lip prevents the possible fracture of the springs by limiting the side play. The packing consists of four white-metal blocks *Q*, each contained in a gun-metal case *R*, which are pressed directly on to the rod by spiral springs *B*. The blocks are fitted in pairs on opposite sides of the rod, one pair above and at right angles to the other. Each pair works between gun-metal guide blocks *N*, which are pressed in by spiral springs. The packing and guide blocks, with their springs, are contained in two nearly similar casings *P*, and dowel pins are fitted to prevent their rotation relatively to each other. A ball-and-socket joint is made between *K* and the casing. Some steam leaks through the upper packing, and can fill the space about the lower blocks, as access is allowed through a clearance space (about $\frac{1}{32}$-inch) at the lip

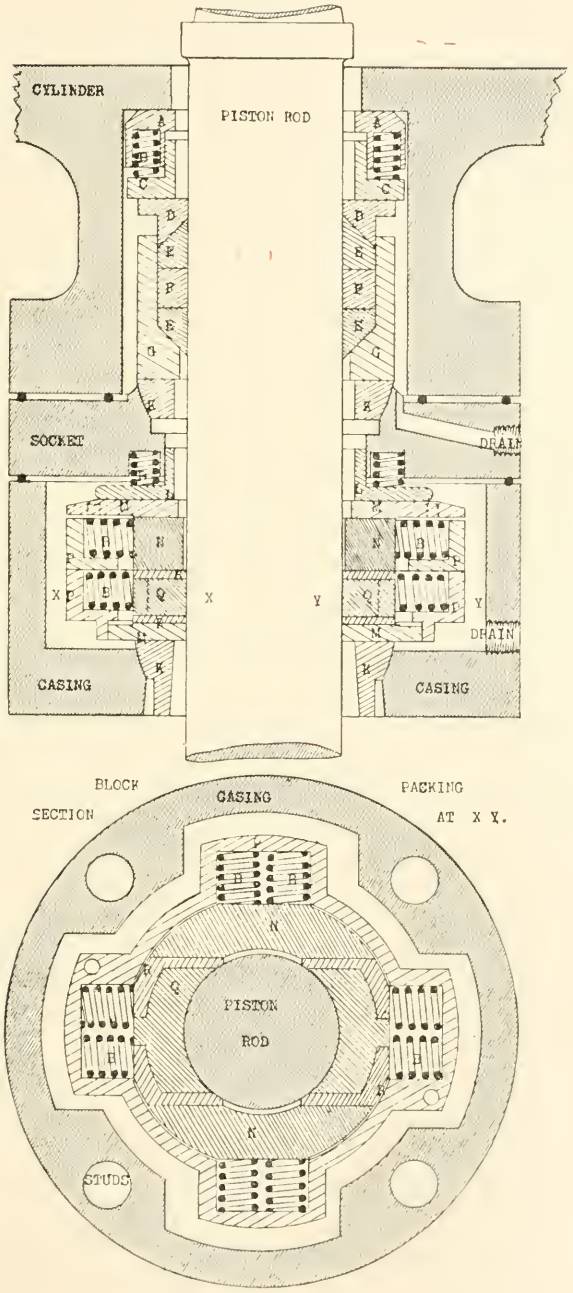


Fig. 163.—U.S. Piston-rod Packing.

of *L*. This pressure assists the springs to press the packing blocks against the rod.

The action of the coned packing has already been described. It is seldom entirely steam-tight, and a second packing is generally fitted to save the water which dribbles past it. With soft packings and many metallic packings, tightness is only obtained by hand screwing up, which produces an unknown pressure on the rubbing surface of the rod. This pressure is constant at all parts of the stroke, and produces an unnecessary frictional resistance.

On the contrary, the springs behind the packing blocks *Q* exert a pressure of about $1\frac{1}{2}$ lb. on each square inch of rubbing surface. Each block embraces about one-third the circumference of the rod, making in all $1\frac{1}{3}$ times the circumference. The depth of the packing is from $1\frac{1}{4}$ to $1\frac{1}{2}$ inches. The pressure exerted by the block packing on the rod varies with the pressure obtaining access from the cylinder to the space between the casing *P* and the blocks, and is in consequence never greater than this amount *plus* the definite pressure exerted by the springs; in other words, the packing is *steam-setting*, and the frictional resistance is a minimum.

The ball-and-socket joints allow a moderate lateral play of the rod, which is a defect common to all rods either from original mal-alignment or from wear. Space is allowed throughout for this lateral play, and when the rod is made with a collar under the piston the space is conveniently used for passing the collar through it. Stronger vertical springs are fitted for use with a vacuum; 16 lb. pressure per square inch is allowed for the L.P. cylinder, and prevents the air passing through the socket joints into the cylinder. The joints of the socket and cylinder stuffing-box are sometimes made with wire-ring joints, as shown in the sketch.

Packing of the block type has been working for many years on land and locomotive engines without any outlay on repairs. It has all the elements of a successful packing, and efficiency can be obtained with it when properly fitted and adjusted.

Slide-valve rod packing is generally of the same description as piston-rod packing. A good white-metal packing is composed of 79 per cent zinc and 7 per cent each of copper, tin and lead.

CHAPTER XVIII

THE CONNECTING RODS AND SHAFTING

Piston Rod.—The connection of the piston with the piston rod has already been described. The rod itself is now generally ground, after turning, to produce perfect parallelism. The connection of the lower or outer end of the piston rod with the connecting rod is made in two ways, as shown in Fig. 164.

1. The top end of the connecting rod is forked, and a single gudgeon pin is fixed through the two ends of the fork. A bearing is fitted in the end of the piston rod, as shown in Fig. 165, and embraces the gudgeon pin.

2. The lower end of the rod is made with a coned end, similar to the one fitted for the reception of the piston, and this cone fits into a cross piece, or *crosshead*, the ends of which form twin gudgeon pins. Bearings are fitted to the top ends of the fork, and these embrace the gudgeon pins, as shown in Fig. 168.

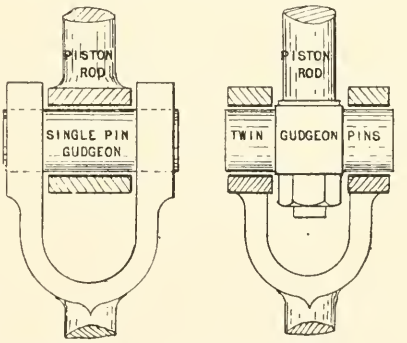


FIG. 164.—Gudgeon Connections.

The pressure exerted on the piston is communicated to the rod, which is alternately in compression and tension. By considering it as a column under compression, the size required generally covers that necessary for the tensile stress. From the piston rod the pressure is transferred to the connecting rod, and so to the crank. The piston and connecting rods are not always in line, and a proportion of the pressure is transferred to a guide, as explained in Chapter XVI. The end of the rod is fitted to transfer the pressure to the guide, and is thus relieved of the bending action to which it is otherwise subjected.

Guide.—The back column is made suitably to receive the guide, or shoe, in which the slipper attached to the end of the piston rod moves. The mutual action of the guide and slipper prevents any bending of the piston rod, which is due to the obliquity of the connecting rod. The guide is generally made separately from the column to which it is bolted, as shown in Figs. 142 and 167. A space left between the column and the guide is made water-tight, and used to keep the ahead rubbing surface cool by a continuous circulation of water through it, preferably in an upward direction. The outlet for the circulating water is generally led into the crank pit, so as to be visible from the starting platform. In mercantile vessels the circulating water is frequently led overboard, above the water-line. Sufficient head of pressure is obtained for this purpose by taking the supply from the main circulating-pump discharge.

A fairly accurate measure of the heat and work wasted in friction at the guide can be obtained by observing the rise of temperature of the outlet water and the weight used in a certain time. Hand-holes are provided behind the guides, so that the temperature of the back of the rubbing surface can be felt while the engines are working.

The astern guide surface is also made separately from the column, and generally from the ahead surface, one strip on each side of the slipper, and these are bolted on to the front of the ahead guide, as shown in Figs. 142 and 167. The area of the astern surface is about three-fourths that of the ahead surface, and is sufficient for ordinary practice. The guides for both ahead and astern working are made of cast iron or steel, and suitable grooves are made in them for distributing the lubrication over the rubbing surfaces.

In some cases, generally in mercantile practice, the front and back columns and guides are identical. A special type of crosshead is then required, and one used for this purpose is generally fitted for twin gudgeon pins. The double guides, as described, are heavier, and when it is considered that a sea-going ship is seldom required to go astern for long periods, the open-fronted engine appears to be suitable and preferable.

Slipper.—For large engines the slipper is usually a separate fitting, because it is difficult to make in one piece with the crosshead, and because it also permits adjustment for wear. Various types, all very similar, are shown in Figs. 165, 166, 167.

In Fig. 165 a horizontal dovetailed groove is cut in the slipper *EE*, which fits tightly over a fillet, *D*, on the crosshead ; and the two

parts are held together by sunk-headed screws, which are secured from slacking back by the set screws shown partly embedded in the heads of the screws. The ahead rubbing surface of white metal is

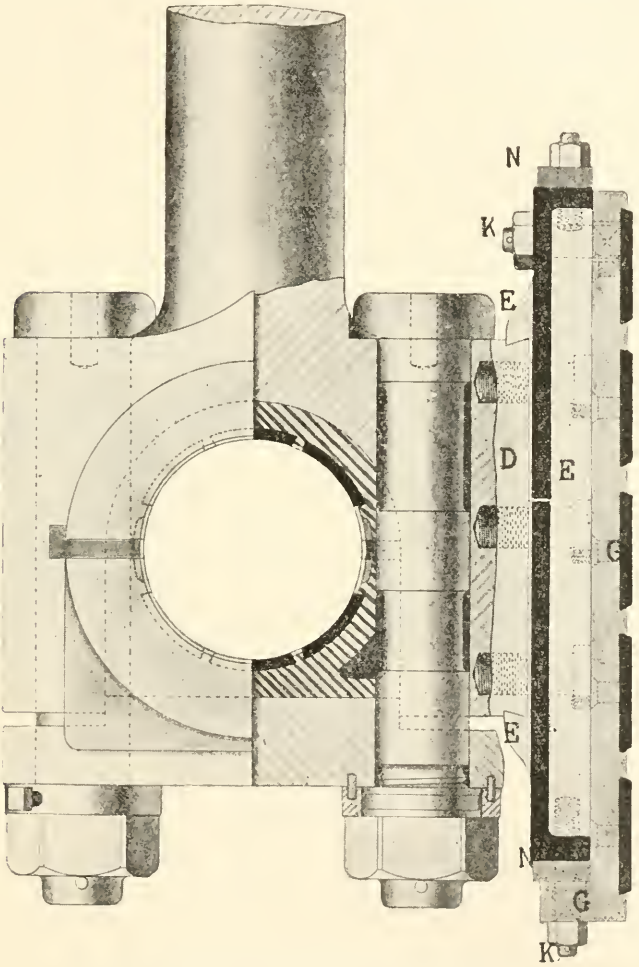


FIG. 165.—Slipper and Single Gudgeon-pin Bearing.

contained in a gun-metal liner *G*, which is removable without disturbing any important parts of the machinery. The liner *G* can be slid upwards into its place, and when the surface is worn, a thin sheet liner is inserted between it and the slipper *E*, to restore the rod to its alignment with

the axis of the piston and cylinder. When in place the liner is held by studs K (fitted with square necks, collars, and the usual split pins, to prevent slacking), and the holes in the liner are elongated, to allow for adjustment. These studs are not quite safe under the tensile stress to which they are subject, and additional studs K are fitted through the slipper and liner at the top end, with the usual nuts and stops

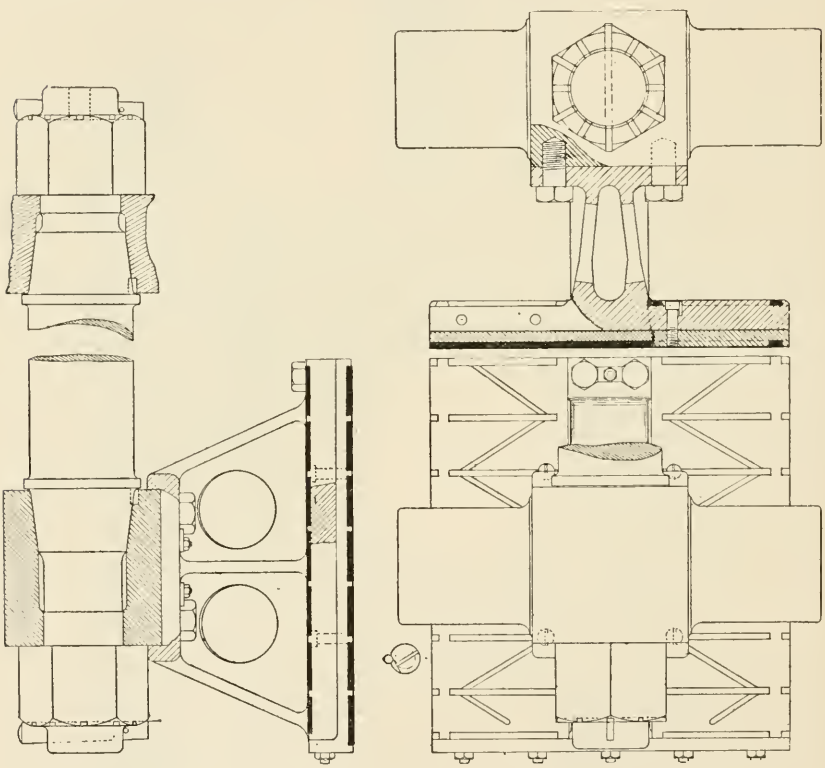


Fig. 166.—Crosshead and Slipper.

to prevent their slacking off. The white metal is dovetailed into the containing liner, and cross spaces are left in it to retain the oil on the rubbing surfaces, and to distribute it over the surface evenly.

The white metal which forms the astern rubbing surface is generally dovetailed into the slipper without any containing gun-metal liner, as shown in Fig. 167. But in a few cases the astern rubbing surfaces of the slipper are detachable (as shown in Fig. 165), made of white

metal, without containing pieces, and inserted from the top and bottom ends of the slipper. Flanges above and below the four strips shown turn inwards, and next these flanges gun-metal strips *N* are fitted to receive the nuts of studs fitted through the strips and flanges into the ends of the slipper.

In good practice the recesses for the white metal are machine

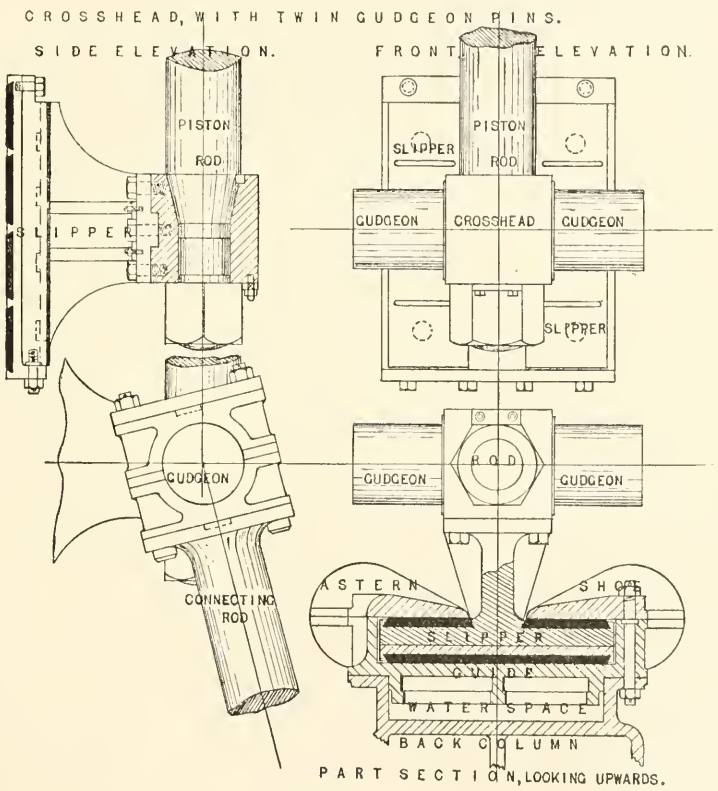


FIG. 167.—Twin Gudgeon Pins and Slipper.

cut, and the white metal, after it is run into them, is machined to shape, and the recesses for lubrication are machined out. White metal is usually fitted for both ahead and astern surfaces of the slipper.

Under ordinary conditions the wear on the astern surface is inappreciable, and therefore the rod tends to retain its alignment when going astern ; but if the extra liner, instead of widening the difference

between the two rubbing surfaces, merely sets the rod end *away from the ahead surface*, it at the same time pushes the astern surface of the slipper away from its guiding surface, and produces mal-alignment. For this reason the adjustable liner should always be fitted between the ahead and the astern rubbing surfaces of the slipper, and *not* between the crosshead and slipper.

Connecting Rod.—The connecting rod is made of forged steel, and there are two principal types.

With the inside or *single gudgeon-pin* type (Fig. 165), if a collar is fitted on the piston rod under the piston, it may necessitate the neck ring, stuffing glands, and other fittings embracing the piston rod (which are of smaller internal diameter than the external diameter of the collar) being cut in segments. This conduces to leakage at the glands, unless special precautions are taken, and the result is not so satisfactory a mechanical fitting as the *uncut* glands, etc. The forging and machining of the piston-rod end is also more cumbersome than the outside or *twin gudgeon-pin* arrangement (Fig. 168), now frequently adopted. It is so proportioned that the connecting rod oscillates just clear of the piston-rod nut at the fork end. Two adjustable bearings are fitted, one to each of the twin gudgeon pins, and properly secured to each fork end. In this arrangement it will be noticed that the *bearings* are carried by the connecting rod, which is the reverse of the previous type, in which the *pin* is carried by the connecting rod.

The length of the connecting rod from centre to centre of its end bearings is almost invariably made equal to four cranks—that is, to twice the length of stroke of the piston. Its minimum diameter is calculated by considering it a pillar in compression, which generally covers its necessary strength considered as a rod in tension. A bending moment due to the centrifugal action of the rod has also to be considered, and to meet this, the rod is tapered, with the larger diameter at the crank end. In very slow-moving engines the rod was made with its largest diameter at about its mid-length, but with the quicker-moving engines of the present day the largest diameter has gradually fallen to the part immediately above the crank.

In some of the older ships, where gun metal was used for the top-end rubbing surfaces, the gudgeon pin was made of wrought iron, case-hardened on its surface ; but with the white-metal linings, now almost universally adopted, mild steel pins are used. The twin pins are forged in one piece with the crosshead. The single pin is shrunk into the holes in the fork, and set screws are screwed in at the junction to

prevent the pin turning; when worn oval it can be trued up by filing or lapping, and is sometimes turned round a quarter-turn so as to bring the oval the opposite way.

Connecting-rod Bearings.—Although there is some considerable

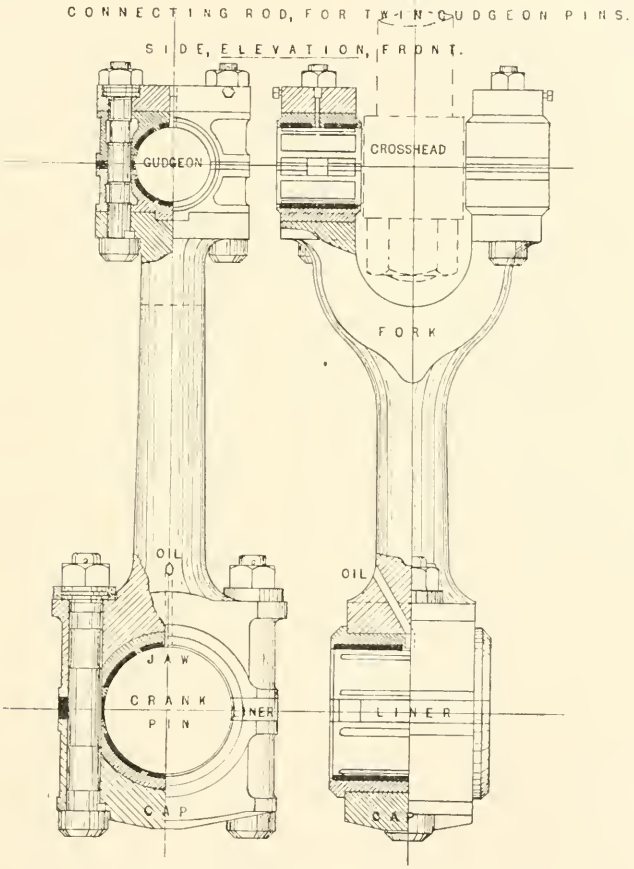


FIG. 168.—Connecting Rod for Twin Gudgeon Pins.

variation in outward form and shape, the same general principles are apparent in all connecting-rod brasses. They are made in two parts, upper and lower; and for convenience the bolts at the top end point downwards when a single gudgeon pin is used, but for twin pins and all crank-head brasses the nuts are conveniently fitted on top. The

outer or cap brass is stiffened by a steel cap (Fig. 168), and liners are fitted between the two parts, so that by reducing their thickness the brasses can be brought together, and thus embrace the pin more closely. The diametrical working clearance between the pin and the brasses should be about 0·1 per cent of the diameter of the pin, but much depends on the perfection of workmanship and the alignment. (See also Chapter XXXVI.)

All the bearings are now lined with white metal in strips, which are fitted longitudinally to the pin ; but the upper and lower edges are usually made of the shape shown in Fig. 435. This arrangement conduces to a better distribution and retention of the lubricant. The white-metal lining extends over about two-thirds of the semicircumference at the top and bottom of a pair of brasses, with the clearance at the distance pieces (liners).

In the more recent ships an *oil-containing strip* of white metal extends through the brasses and liners, so as to completely embrace the pin as near each end as possible ; this is to prevent the oil working out of the bearings too quickly. A strip is shown in the part section of the crank-head brasses in Fig. 168, and strips are also used for main-bearing and gudgeon brasses.

The *connecting-rod bolts* are made of steel, and now generally fitted with wrought-iron nuts ; the bolts are made with stretching lengths just inside the nuts, and stops are fitted under the heads, to prevent them turning in their holes. A little consideration shows that the bolts in the connecting rod and main bearings are only subject to a tensile stress, and are never in compression.

For the proper adjustment of the connecting-rod brasses, the opposite end of the connecting rod should be disconnected and the rod swung through the working angle, and adjusted until it swings fairly in line. This swinging adjustment is particularly necessary with twin gudgeon pins. For the lower or crank-head brasses the rod should be swung at various positions of the crank to completely test the adjustment.

Main Bearing.—The crank shaft rotates in the main bearings, and one is fitted on each side of each crank as closely as possible, to reduce the bending moment on the shaft, as shown in Fig. 172.

The main-bearing brasses are white-metal lined, and the oil-containing strips are fitted near the ends of the brasses, completely embracing the shaft, as in the crank-head brasses. The lower brass is made semicircular (Fig. 169), so that when the steel cap, top brass, and liners

are removed, it can be turned round on the shaft until it is bottom upwards, and can then be lifted away for examination and repair; thus without lifting the crank shaft the bottom main-bearing brasses can be removed and refitted. A hand-hole is made in the cap, so that the top brass can be felt when the engines are working. Steel bolts fitted with a collar have been substituted for studs securing the main bearings in comparatively recent years. Studs have an unpleasant habit of slacking back which cannot be easily detected. The lower end of the bolt is fitted with a wrought-iron nut, generally square,

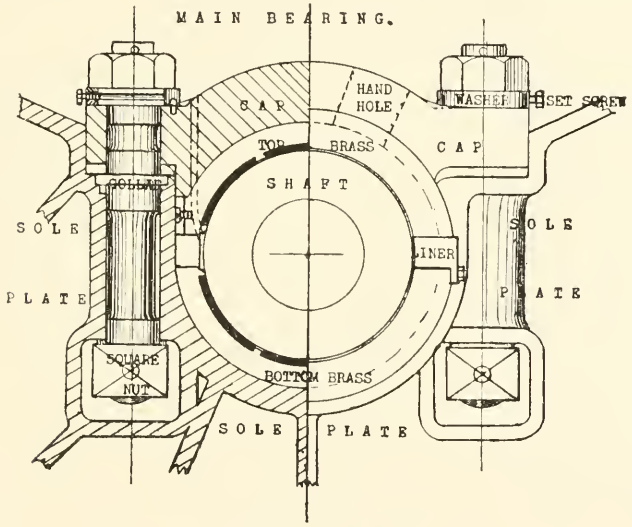


FIG. 169.—Main Bearing.

secured by a pin through it; the bolt is prevented from turning by a stop under the collar; a stretching length is fitted inside the cap, as shown, and the usual wrought-iron spigoted nut, washer, dowel pins, set screw, guard plate, and split pin are fitted.

In addition to the lubricator fitted for the top of the top brass, another is now fitted, and conducts the oil to the lubricating hole, shown in dotted lines in the figure, leading to the middle space about the liner for the independent lubrication of the bottom rubbing surface.

Centrifugal Lubricator for Crank-pin Bearing.—The crank pin is usually made hollow, and is then used for lubrication, as shown in Fig. 170. The oil runs down a small pipe *A* into a double groove *B*, which is fitted on to the side or web of the crank, and revolves with it.

The centrifugal force given to the oil by the rotation of the shaft drives it into the pipe C, leading into a pipe D, or inclined hole in the pin E, and thence through a hole F drilled through the pin and leading to the bearing surface. Two such double grooves are fitted to each large crank, one on each side. It has been found that when water is used on the main bearings it finds its way into the crank bearing, and to prevent this happening, a lip G is now fitted, as shown in the figure, and also an enlargement, which throws the water off. This lip allows dirt to fall into the double groove, and over it a plate K is fitted on each main bearing to protect it. In all cases a wire-gauze screen is fitted at some part of the outlet pipe into the crank, generally at the joint

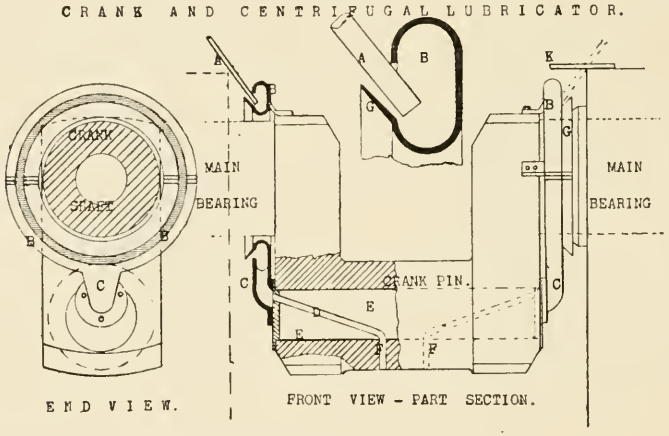


Fig. 170.—Centrifugal Lubricator and Crank.

between the outlet pipe C from the double groove and the end of the crank. The centrifugal lubricator must be made in two semicircular parts, so as to get it in place. Lugs are fitted to it, and make the connection with the crank. Oil-ways are cut as necessary in the crank-pin surface.

Forced Lubrication.—In several modern ships fitted with reciprocating engines a system of lubrication under pressure has been adopted with satisfactory results. The crank pits are arranged (see Fig. 171) so as to form a well for the oil dripping from the bearings, and which drains naturally into the oil-pump suction, which is fitted as a filter. The oil is cooled on its way to the pump by means of a cooler—the water suction for which is shown in the sketch. The cooler is of the surface type, and the oil and water do not mix. One engine generally

performs the separate duties of pumping water through the cooler and of pumping oil under pressure into the bearings. Oil under pressure is forced by the pump through pipes, shown by thick lines in the plan, into the main bearings, from which it flows through holes in the shaft into the inner space of the shaft. From this inner space the oil is forced through small holes into the interior of the crank-pins and so to the bearing surfaces of the crank bearing; a small hole leads to each eccentric bearing surface, and in some instances a hole in the

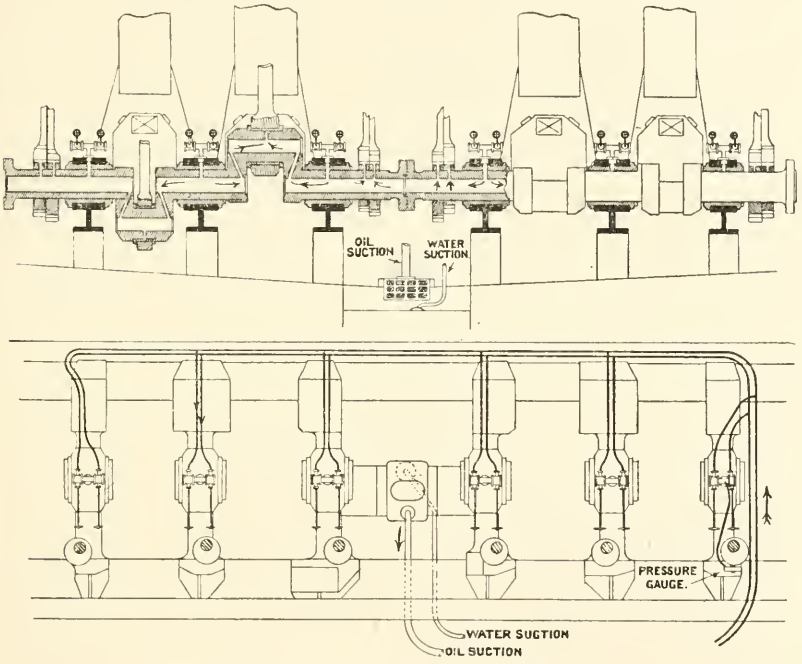


FIG. 171.—Forced Lubrication System.

interior of the connecting rod conveys oil to the crosshead bearing. For forced lubrication it is necessary to use pure mineral oil only, because olive oil, although a superior lubricant, tends to saponify and choke the filter, and this tendency is increased by any mixture with either fresh or sea water. A pressure-gauge shows the pressure in the pump delivery; the pressure should generally be regulated to suit the power of the engine; in round numbers 10 lb. per square inch should be allowed for each one-fifth of the full power, but in this as in other matters of lubrication practical experience is the best guide,

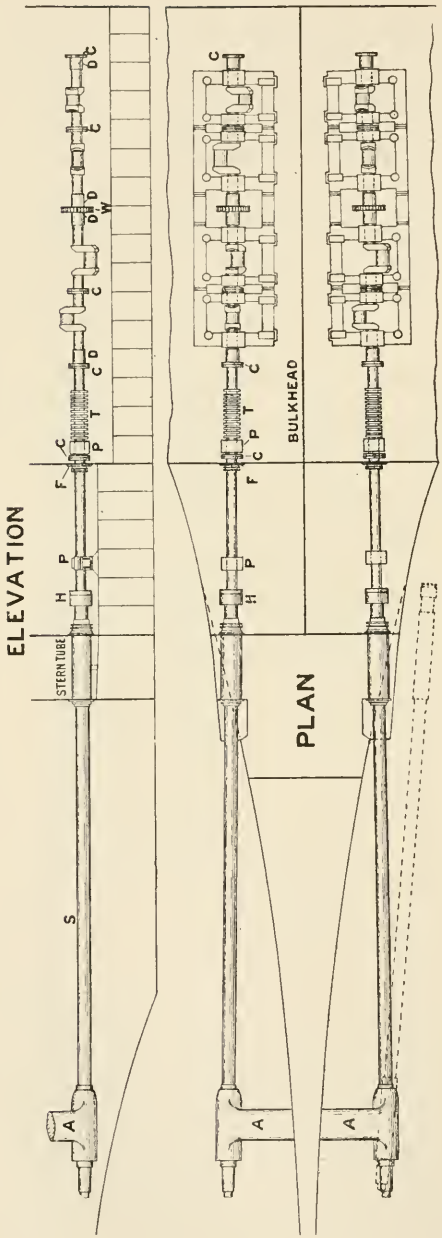


FIG. 172.—Shafting and Bearings.

and what may be successful and satisfactory in one case may not necessarily be so in another engine room. Oil-containing strips are necessary to a successful system of forced lubrication, and careful fitting and adjustment of the bearings helps considerably. The figure shows a duplicate system of supply pipes.

Propeller Shafting.—In Fig. 172 a plan and elevation of the shafting of a twin-screw ship is shown in general outline, with the positions of the various fittings and bearings. The arrangement of each engine and shafting is similar, but fitted on opposite sides of the keel. The two engine rooms are divided by a middle-line bulkhead, and each engine is made so as to be entirely independent of the other.

The shafting shown is for the four-crank, four-cylinder, three-stage expansion type, with two L.P. cylinders. The crank shaft is made in two or four lengths, and to some extent these are interchangeable, and for this reason a coupling *C* is frequently fitted at the

forward end of the forward length. The adoption of varying angles between the cranks, as used in the Yarrow-Schlick-Tweedy and other balancing systems, sometimes prohibits entire changeability, which is, however, always desirable. At the positions for the eccentrics *D* the shafting is enlarged, so that the cutting of the keyways for securing the eccentric sheaves on the shaft shall not weaken it locally. For each coupling there is a fillet at one end fitting accurately into an opposite socket; this is to maintain a perfect alignment of the shaft. The coupling bolts are a driving fit into their holes, and frequently, to shorten the length of the shaft, and so bring the bearings closer together, the heads of the bolts are coned, and wholly or partially sunk into the coupling flange. Cone-headed bolts should not be used for couplings subject to propeller or end thrust. The sketch shows the outline of the sole plate into which the main bearings are fitted.

Just abaft the after coupling the thrust shaft *T* is fitted, and next abaft is a plummer block *P* frequently incorporated with the thrust seating. Then comes another coupling at the after end of the thrust shaft for coupling with the intermediate shaft, and to this coupling the turning wheel is attached, if there is sufficient room for it; but if not, then it is fitted at *W*, or as convenient. The intermediate shaft projects into the engine room from the shaft alley, and a hole sufficiently large for its withdrawal is cut in the bulkhead at the fore end of the alley. This hole is covered with two semicircular plates, bolted on to the bulkhead and fitted with a packing gland *F* so that the bulkhead can be made water-tight. Access from the engine room to the shaft alley is provided by means of a water-tight door, which can be closed or opened both from the engine room and from some point above the water-line.

The intermediate shaft is supported by plummer blocks *P*, and at its after end is provided with a coupling suitable to the loose coupling *H* at the fore end of the stern shaft *S*. In some smaller ships the shaft alleys form part of the engine rooms; but as the length of ships is increasing, they now generally form separate compartments. In many of the older naval vessels there was an outside coupling abaft the loose coupling, which was sometimes protected by a partially removable tabular casing, the after end of which was secured to the stern, or "A," bracket. A better flow of water to the screw is obtained in later vessels by omitting the external casing and outside coupling; a coned piece is then fitted in front of the "A" bracket, rising from the shaft, to give an uninterrupted flow of water to the screw.

Plummer Block.—The main bearings support only the crank shaft, and the weight of the propelling shaft must be supported at suitable distances to maintain proper alignment. Unlike the main bearings, which have to take a stress upwards as well as downwards, the plummer block has only the dead weight of the shaft to support; only the lower part of the block is therefore lined as a bearing with white metal, and the upper part is merely a cap, and does not come in contact with the shaft. The cap is used to keep dust and dirt out of the bearing, and is made with the necessary fittings for holding the lubricator and water-service pipes. In Fig. 174 a plummer block is shown incorporated with the thrust block at its after end. If the thrust block is not sufficiently close to the after-crank bearing, another plummer block is fitted at the fore end of the thrust block. It is

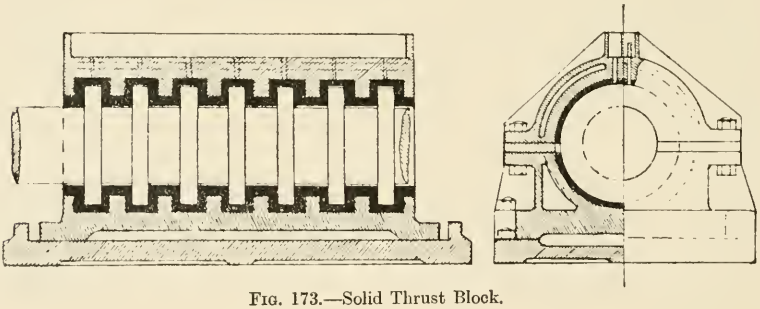


FIG. 173.—Solid Thrust Block.

important that the latter should not be burdened with the additional duty of supporting the shaft. The plummer blocks, as independent fittings, are supported on two or more frames of the ship, so as to distribute the weight of the shaft.

Thrust Block.—When the propeller is rotated it exerts either a pull or a push on the propeller shafting, by which the ship is moved. Some means must be provided for transferring this force to the ship itself, or thrust would be produced on the engine and framing. With screw propellers this force is transferred through the medium of the thrust block, of which a simple type is shown in Fig. 173. On the shaft several collars are formed, and these fit into recesses accurately made for them in the block. This block is white-metal lined (shown in black section in figure), and there is radial clearance (about $\frac{1}{8}$-inch in large engines) between the recesses and the collars, so that the weight of the shaft shall not rest on the block, which is provided to take the thrust only. The top and bottom parts of the block are

generally made hollow, and water can be circulated through the spaces so formed to keep the bearings cool. A large proportion of the power developed must be dissipated in friction at the collars, and for this reason water service is usually necessary to carry away the heat thus generated. The block is securely bolted down to a seating plate, which rests on several frames of the ship, to which it is secured, by means of longitudinal tie beams and girders, so as to distribute the thrust pressure over a sufficiency of the plating and framing of the ship. This block can be adjusted to a moderate extent by means of set screws and taper keys fitted between it and the raised ends of the seating. The bolt holes at the bottom of the block are elongated for this purpose, and generally filling pieces are fitted alongside the bolts after the block is properly adjusted, to prevent it shifting on its seating.

Horseshoe Thrust Block.—For large ships and great powers a more elaborate and also useful type of block was adopted, and is shown in Fig. 174. Generally this block has an arrangement by which the water used for cooling the bearing surfaces is not allowed to come into actual contact with them.

Five sketches are shown. The left-hand top one shows an end view just in front of one of the thrust-shaft collars ; the right-hand top shows an elevation in part section ; the lower right shows a plan with the shoes removed and with the thrust bars and nuts for adjustment ; the mid left shows a section across one of the horseshoe rings with the space for the internal water service and its connections ; while the bottom left shows a plan of one of the rings.

The advantage of the separate rings or horseshoes is that each ring can be adjusted or renewed independently, and this can be done very quickly ; less spare gear is required, and a fair speed can be maintained with only a few rings in effective operation. With all the rings in one containing casting, it is probable that they would all give way together, and the ship might become disabled for a long time. With separate rings, it is possible to renew them one at a time while the engines are at work.

The centre of pressure on the rings is above the centre of the shaft, and it is arranged that the centre lines of the thrust bars E are level with the centre of pressure. There is, in all thrust blocks, a difficulty in getting the lubricating oil to remain on the rubbing surfaces, and a special cutting of the oil grooves on the rings is the result, as shown in Fig. 174. It will be noticed that there is an additional hole conducting the oil to the lower part of the shoe, and that the rotation of

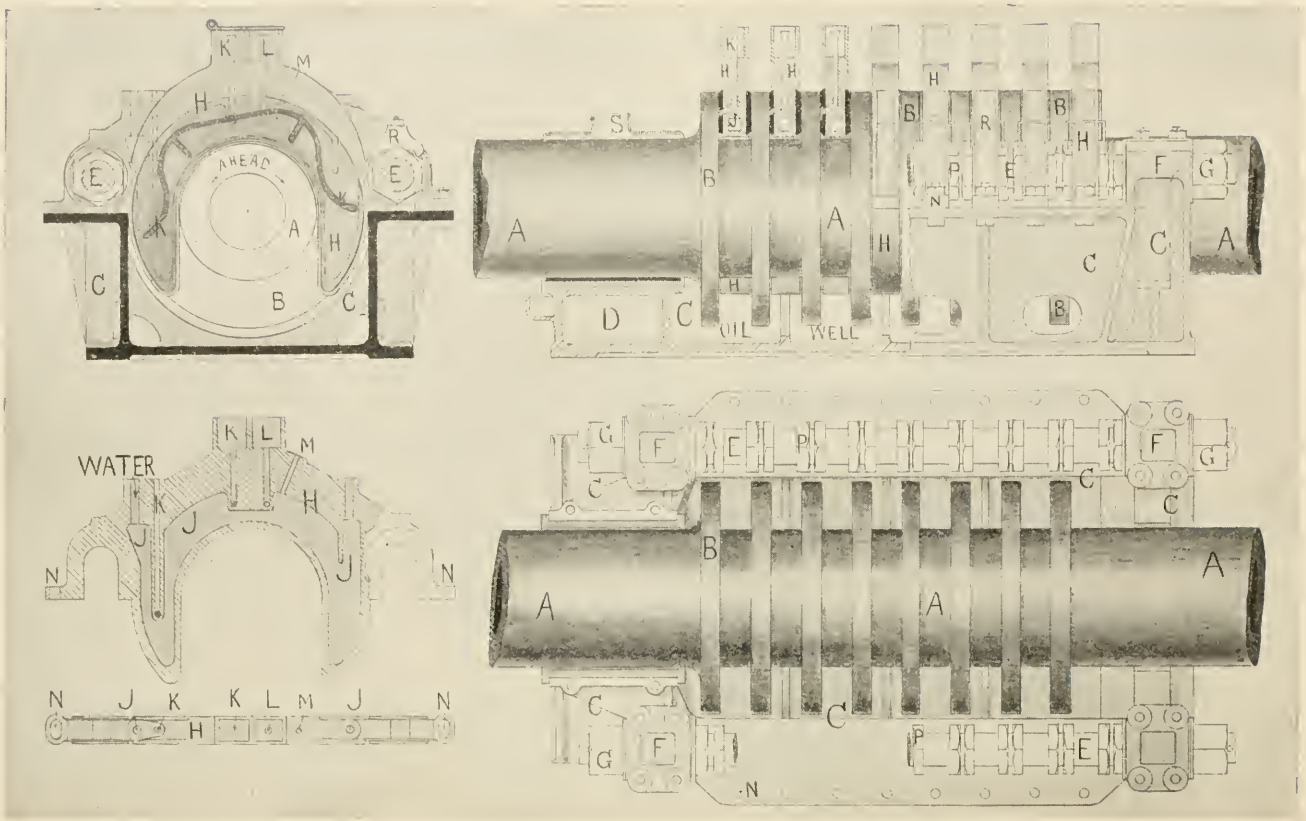


Fig. 174.—Horseshoe Thrust Block, with Internal Water Circulation.

the shaft in the ahead direction tends to take the lubricant with it, both on its surface and in the groove which assists in supplying it. The oil that escapes from the surfaces fall into the bath, or well, at the bottom, and is again drawn up by the collars, which are partially immersed in it. A single lubricating hole at the top of each collar serves for the astern surface.

REFERENCES TO FIG. 174.

A, Steel thrust shaft, generally hollow.
B, Collars on shaft *A*.
C, Cast-steel frame, securely bolted down to ship's framing, which takes the propeller thrust.
D, Part of *C*, which forms the lower part of the plummer block at after end of the thrust shaft. If the thrust block is not close enough to the after-crank bearing, a plummer block is also fitted at the forward end.
E, Steel-screwed thrust bars, secured to framing *C*.
F, Steel removable caps, fitted over ends of thrust bars to secure them down to the framing *C*.
G, Wrought-iron nuts at end of bars, used to take the thrust, and also for adjustment.
H, Horseshoe thrust rings, lined with white metal on both sides next thrust collars on shaft.
J, Hollow space in each shoe, through which water is circulated to keep the bearings cool.
K, Oil ways, and oil groove in white-metal lining for **ahead** thrust.
L, Oil way for **astern** thrust.
M, Air outlet, fitted with a cock, from circulating water.
N, Bolt holes for holding down each shoe to frame *C*. Holes in shoes are elongated, to allow for adjustment fore and aft.
P, Phosphor-bronze check nuts, each side of each shoe, on screwed thrust bars *E*, for adjustment of each shoe independently.
R, Guards to prevent nuts slacking back.

The exclusion of the sea-water service from actual contact with the rubbing surfaces conduces to maintaining great and continuous efficiency. Water produces corrosion, then a roughness, which produces a great frictional loss. Good and efficient lubrication prevents rubbing surfaces coming into metallic contact by interposing a film of oil between them ; this is the object of all lubricating arrangements.

In several ships an improved method of lubricating the thrust collar surfaces has been adopted. Instead of, or in addition to, cutting grooves in the white metal of the horseshoe collars, other grooves, about $\frac{1}{4}$-inch wide and $\frac{1}{8}$-inch deep, *well rounded at the cutting edges*, and extending to within 1 inch of the edges of the collars, are cut, nearly tangentially to the shaft, in the faces of the shaft collars, as shown in Fig. 175. The collars are always partially immersed in oil contained in the well formed by the lower part of the block. Some oil is caught up on the bearing surfaces of the collars, and instead of being thrown off by centrifugal action is retained by the grooves and carried from them to the surfaces. A very efficient lubrication can be obtained in this way with greater simplicity and less chance of the grooves choking than when the latter are cut in the white metal. The grooves must necessarily be cut in the opposite directions for *ahead* and *astern* surfaces respectively.

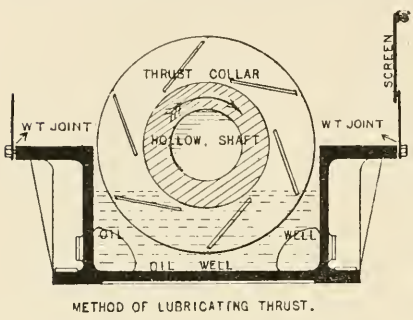


Fig. 175.—Oil Grooves in Shaft Collars.

Michell Oil Film Lubrication.—All the previously mentioned methods of taking the propeller thrust have been eclipsed by the invention of the Michell Thrust Block which has only a single collar, as shown in Figs. 176, 177, and 267, and its film system of lubrication.

Mr. A. G. M. Michell found by experiment that when two flat surfaces ride one over the other it is necessary that one of the surfaces must be free to take up a slight inclination to the other to maintain an oil film between them. This is the essential condition in a perfectly lubricated thrust block, otherwise the oil escapes more quickly than it can enter, until none is left and metallic contact takes place with consequent friction and wear. By adopting film lubrication the Michell block is capable of carrying a thrust pressure of 500 lb. per square inch compared with a maximum of 50 lb. carried by the older systems of parallel surfaces.

In the Michell system the fixed surface is divided into a number of segments, each of which is pivoted to enable it to take up the necessary angle to the moving surface and thus to ensure the maintenance of a pressure oil film. This angle varies according to the pressure, speed,

and kind of oil used, and the segments are accordingly arranged to adjust themselves to suit the varying conditions.

Fig. 176 illustrates the methods of pivoting by line or point, and each of these methods is satisfactory in practice. The best line or point on which to pivot the segments or thrust pads is somewhat *behind* the centre, the general practice being at 62 to 66 per cent, which approximately coincides with the position of maximum thrust between

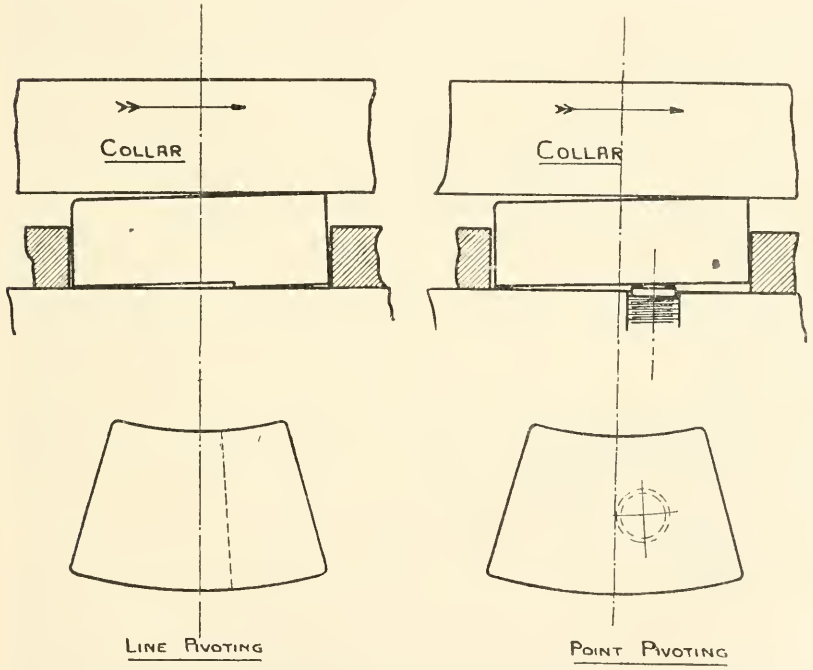


FIG. 176.—Michell Oil Film Lubrication.

the two surfaces. From oil film experiments and practice, it is found that the length of the pad should not exceed its width, or radial length, and in practice these two dimensions are made approximately equal.

Fig. 177 illustrates a type of Michell Thrust Block made by Messrs. Broom & Wade of High Wycombe for large mercantile and passenger vessels, whether driven by reciprocating or turbine engines. The thrust pads *A* in this instance are carried in spherical blocks *B*, which bear against spherical rings *C*, in the block casing *D*, which thus enables the pads to accommodate themselves to the face of the collar and to

take an equal share of the load. Any slight disturbance of shaft alignment due to disposition of cargo, or to heavy weather (especially destroyers and lightly built vessels), is thus met.

Adjustment is effected through adjusting rings or washers *E* behind the spherical rings, and spare adjusting rings of slightly varying thickness are carried on board for this purpose. Hardened steel pivots *F* are frequently let into the beak of each pad.

The design for naval vessels is very similar to that shown in Fig. 177, and it has enabled very large powers to be carried on a single block, which, compared with old methods, effect a large saving in weight

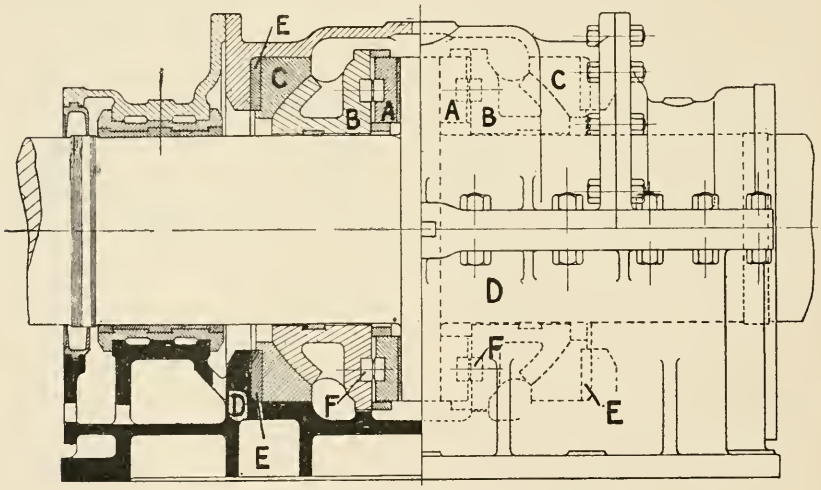


Fig. 177.—Michell Thrust Block and Adjustment.

and space. The Michell thrust may be considered as an almost indispensable adjunct to geared turbines, in which the whole of the propeller thrust must necessarily be taken on the shafting. Many modern turbines are fitted with Michell blocks for taking the turbine thrust (See Fig. 267.)

For lubrication of the Michell Thrust Block it is advisable to use the thinnest oil that will prevent metallic contact at the particular speed of the bearings, because the friction increases with viscosity of the oil used. For blocks where the rubbing speed is comparatively low, a satisfactory method of lubrication is to run the block in oil, kept cool if necessary by water circulation. For the larger blocks and those running at high speeds the most reliable lubrication is obtained by

circulating oil through the bearing, and an oil cooler is generally fitted in the forced lubrication system.

Existing multi-collar thrust blocks can usually be altered to obtain increased reliability on the Michell system, especially those of the horseshoe type adjusted by side screws.

An excellent description of experiments with the Michell system and its practical application is given in a paper read before the Institution of Naval Architects, April 10, 1919, by Dr. Hamilton Gibson, O.B.E., and published in *Engineering*, June 13, 1919. The relative coefficients of friction are given as 0·0015 for rocking pads, and 0·03 for flat thrusts of the old type which thus appears to be twenty times as great.

Messrs. Broom & Wade tested some $7\frac{1}{2}$-inch bearings of the Michell type for the Admiralty with the following results:

Load	20,000 lb.
Pressure on pads . . .	400 lb. per sq. inch.
Speed	460 revs. per minute.
Coefficient of friction . .	0·0015.

These blocks, or rocking pads, are usually designed to carry a pressure of from 200 to 300 lb. per square inch, but as experience and confidence is gained in their efficiency it is probable that the pressure will be gradually increased up to about 400 lb. for geared turbine sets where the speed is moderate.

Messrs. Broom & Wade make a number of special applications of the Michell system in place of Ball bearings, for centrifugal pumps, fans, worm gears, etc., for which they have patented a swivel collar of simple construction.

Michell Journal Bearing.—This bearing consists of a number of small pad pieces, six are shown in Fig. 178, having white-metal bearing surfaces supporting the journal; these pads are contained in a suitable housing supported in a bearing block. The backs of the pads are so shaped that each pad is free to rock about a line situated a small distance from the geometric centre of the pad towards the trailing edge.

When running, the pad pieces rock about this line and thus the bearing surface does not lie concentric with the surface of the shaft; the space between the two surfaces assuming a wedge form with the thick end of the wedge near the leading edge of the pad. The lubricating oil enters at this leading edge and as it passes over the surface the oil pressure is greatly increased due to the wedge formation and

forms a film between the two metallic surfaces, which is maintained so long as the shaft is in motion.

In the case shown in the illustration, the bearings were made to replace the existing plain journal bearings and the length of the bearings was reduced from 5 to 2 inches. This increased the pressure on the projected areas from 100 lb. to 352 lb. per square inch, which proved quite satisfactory under trial. The housing for the new bearing had to be adapted to suit the existing block, and a portion of the housing was arranged with a plain ordinary bearing kept just clear of the

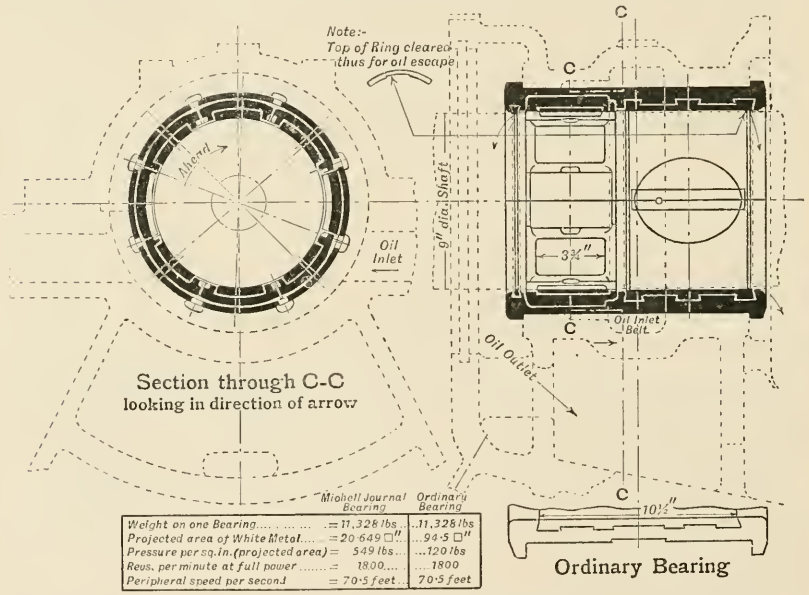

Note:-
Top of Ring cleared thus for oil escape
Ahead
Oil Inlet
9" dia. Shaft
C
C
3¾"
Oil Inlet Belt
Oil Outlet
Section through C-C
looking in direction of arrow
Michell Journal Bearing
Ordinary Bearing
Weight on one Bearing ..... = 11,326 lbs ... 11,326 lbs
Projected area of White Metal .... = 20·649 □". ... 94·5 □"
Pressure per sq. in. (projected area) = 549 lbs. ... 120 lbs
Revs. per minute at full power ..... = 1800 ... 1800
Peripheral speed per second = 70·5 feet ... 70·5 feet
10½"
Ordinary Bearing

Fig. 178.—Michell Journal Bearing.

journal and intended to come into action should the pads fail. The test showed that this bearing took no weight and was therefore unnecessary, and indicated great possibilities of reduction in the length of housing, and consequential reductions of total length of machinery and weight.

Michell Journal and Thrust Block.—A further application of the Michell oil film principle is shown in Fig. 179 of a thrust block combined with a journal bearing for carrying the weight of the shaft. This was fitted in the destroyer *Mackay* (vide *The Engineer*, Aug. 29, 1919) and proved successful under trial conditions. At

full power the temperature of the bearing and thrust never exceeded 95° Fahr.

The tipping pads in a journal bearing must necessarily be adapted for one direction of running only, which in a steam vessel would be *ahead*. As the running *astern* is only for short periods they are efficient for this purpose, and no undue heating was observed in the above-mentioned trials.

For turbine machinery where weight and space are of considerable

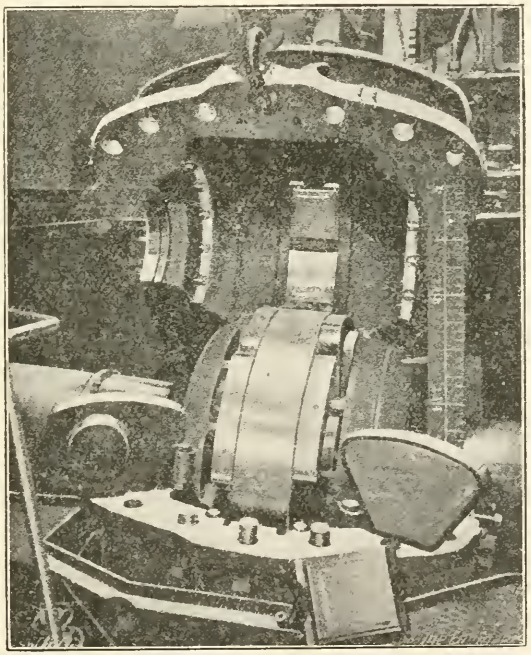


FIG. 179.—Michell Journal and Thrust Block.

importance, the combined arrangement, together with the short Michell journal bearings for the turbine rotors in the typical arrangement, permit an over-all saving in length of machinery space which is claimed to be as high as 20 per cent.

Stern Tube.—The engine is connected by the shafting with the propeller, which is outside the ship, and where the shafting passes through the hull, it is necessary that it should be not only free to rotate the propeller, but also that the water should be excluded from entering the hull. The propeller is fitted well clear of the hull at its

largest transverse section, to allow a free flow of water to it in a manner as little disturbed as possible ; and there is consequently a great length of shafting abaft the aftermost plummer block which requires supporting. This support is given by the stern-tube bearings and a bracket, which is fitted just in front of each screw in twin- or multiple-screw ships (Fig. 180).

Inside the stern bearings, and for rather more than the whole length of the stern tube, the shaft is sheathed with a gun-metal casing, which forms its bearing or rubbing surface, *F*. The sheathing is either forced or shrunk on to the shaft, which it fits closely in the vicinity of the bearings. In many instances this casing is undercut both internally and externally between the bearings. The space between the sheathing and the shaft is filled with a mixture of red lead and linseed oil, forced in by a pump, and left at a pressure of about 30 lb. per square inch. Care is taken to allow the air to escape while the mixture is being pumped in.

The stern tube itself, which forms the support for the stern bearings, is built as part of the hull, and the ship is specially strengthened and stiffened at this part for the purpose (Fig. 180). The inboard end is riveted to the transverse framing, and generally this framing forms a water-tight bulkhead between two compartments of the ship. The bulkhead is stiffened by rings, *C*. and the tube is riveted to the ring *C*, which is generally an angle steel. The after end is of similar construction, but without the packing gland arrangements, *G* and *E*. At the forward and after ends of the tube, cast-iron rings, *D*, are riveted to the transverse framing. These are accurately bored for the reception of a long gun-metal sleeve, which is made with suitable recesses for the bearing strips at each end, and at the inboard or forward end a stuffing box, *E*, is also formed. The bearing strips are made of *lignum vitæ*, and are dovetailed into the sleeve, which should not be allowed to come in rubbing contact with the sheathed shaft and its revolving surface. In nearly all cases, especially for the outer end, the strips are fitted into semicircular gun-metal casings, in order that they can be easily withdrawn for examination or refit, and then screwed studs are fitted to prevent these casings with their contained strips from being carried round with the revolving shaft. The strips in the lower part of each bearing are generally fitted with the grain end on to the shaft, which considerably reduces the wear.

The water outside the ship is allowed free access to the space between the strips and between the revolving and fixed casings ; and

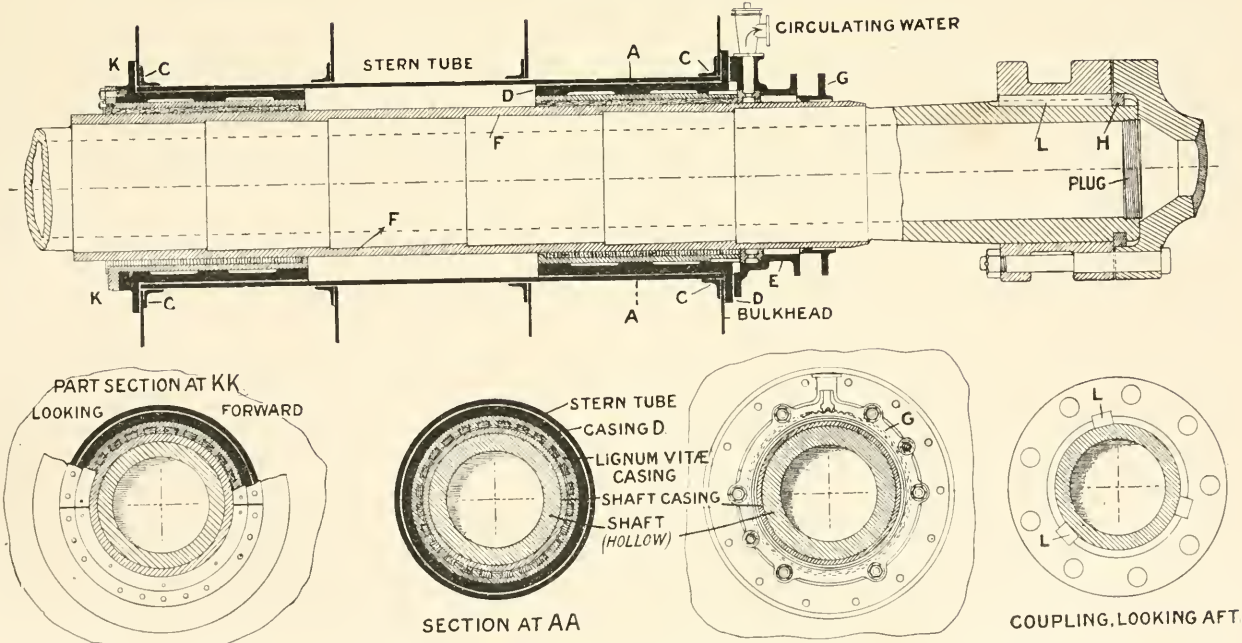


Fig. 180.—Stern Tube, Shaft, and Loose Coupling.

a cock is fitted at the inboard end at the top of the stuffing box, to allow water to circulate when required. To prevent the unnecessary admission of water at the stern-shaft bearing, a packing gland, *G*, is fitted and packed with soft packing, generally gasket well soaked in tallow. The nuts on the gland studs are fitted with cogs, which are geared into a single cogged-ring, to facilitate adjustment by screwing up the gland evenly.

Shaft, or "A," Bracket.—In twin-screw ships it is sometimes necessary to support the shaft outside the ship. This is done by means of a bracket, fitted just in front of the propeller. The upper end of the bracket is generally riveted to the protective deck, where one is fitted which projects outboard for the purpose; the lower end is riveted to a shoe plate, projecting from and forming part of the ship's keel. The bracket is fitted with a *lignum vitæ* bearing in a similar manner to that fitted in the stern bearings, and it is made in two semicircular parts, bodily removable, to allow for the removal of the shaft from the stern tube. The space between the boss of the screw and the bracket is cased in as far as possible (Fig. 369), to prevent ropes or other obstructions being caught up by the revolving shaft and its connections. The casing is preferably made of zinc, and then it serves as a protection against galvanic action; the surface of the rounded part of the bracket is also covered with zinc in many ships.

Stern-shaft Coupling, or Loose Coupling.—The aftermost coupling inside the ship is made in such a way that the shafting inside the stern tube can be withdrawn outboard when required; but when the engine is working astern the withdrawal must be prevented.

A simple type of loose coupling is shown in Fig. 180, and is frequently used. The coupling at the after end of the intermediate shaft is suitably made with a hollow space (shown). This coupling is solid with its shaft. The loose coupling is fitted to the forward end of the stern shaft, and is secured to it by three or four keys, *L*, and therefore revolves with it. This coupling is generally made of wrought iron where a steel shaft is employed, as it is then less liable to set fast. The stern shaft projects beyond its coupling, and into a recess two half-rings, *H*, fit accurately, with no fore-and-aft play. The coupling bolts draw the couplings together after the half-rings are inserted.

When going ahead, the thrust is imparted from the stern shaft partly by the ring and partly by a shoulder formed on the shaft against

which the coupling is butted ; when going astern, the half-rings take the pull on the shaft and transmit it to the thrust block.

When the shaft is hollow, a plug is fitted to the forward end of the stern shaft. In case of fracture, it prevents the entry of water into the ship through the hole in the shafting. In some ships, instead of the half-rings a special capped nut is fitted with a keep to prevent it from slacking back. The thread used on the end of the shaft for the reception of the nut should be of the opposite hand to that of the screw propeller to which the shafting is attached.

Method of Removing the Shafting.—All the shafting inside the ship must be lifted into it and out of it through the hatchways, unless the couplings are separate fittings.

When the shafting outside the ship is of such great length that docking becomes difficult on account of the length of floor required, it is sometimes made in two pieces, joined by an outside coupling just abaft the stern tube and clear of the hull. In big ships the part external to the hull is sometimes encased in a thin-sheet steel tube, which extends as far as the shaft bracket, and is generally secured to it. The internal diameter of this tube casing must be large enough to clear the outside coupling. To remove the outside shaft, it is necessary to first remove the propeller and boss from the end of the shaft and part of the exterior casing mentioned above, then to support the weight of the shaft independently, then to remove the bearing in the stern bracket and take out the coupling bolts ; then by withdrawing the shaft aft a little, the coupling is allowed to fall clear of that on the stern shaft, and the outside shaft can then be drawn foward sufficiently to clear the bracket. After the outer shaft is removed, the stern shaft can be withdrawn outboards from the stern tube. It is necessary to so proportion the lengths of the shafting that any one can be removed or renewed without disturbing any of the permanent fittings. In some cases it is not necessary to first remove the outside length to withdraw the stern shaft, as the former can be taken back a sufficient distance through the bracket bearing.

In single-screw ships sufficient length must be allowed in the dock for the removal of the shafting outboards, and a little consideration shows that this is not very convenient, as it involves a very long dock.

Since about 1900 the outside coupling and the stern-shaft casing have not been fitted in the British Navy. The propeller shafts, although of great length (about 80 feet in the example shown in Fig. 172) can be removed by uncoupling at the loose coupling, removing the

propeller, and, after taking the weight of the shaft on slings, removing the semicircular bearing cases in the " A " bracket. Then, by withdrawing the shaft aft until its fore end is clear of the stern tube, the fore end can be moved away (as shown in dotted lines), and the shaft moved forward until its after end is clear of the " A " bracket. The bearing cases and the hole in the bracket are made sufficiently great in diameter to allow the shaft to be moved out of line.

Stern Bearing Lubrication.—The efficient lubrication of the stern bearing in single-screw vessels, or of the " A " bracket bearings in multi-screw vessels, is of considerable importance and has made good progress in recent years. In Fig. 181 the Ben. R. Vickers &

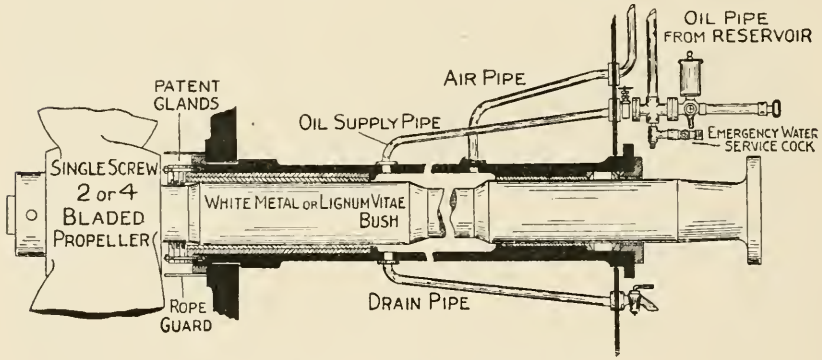


FIG. 181.—Stern-tube Lubrication (B. Vickers & Sons).

Sons system is illustrated and has been fitted with efficient results in many single-screw vessels. The sea end of the bearing is fitted with a special gland consisting of a metal ring revolving with the shaft followed by an elastic felt washer and next the sea by a composition washer. In the sketch two metal rings are shown and one composition washer. The outer washer next the sea is intended to take any surplus of *head* over the internal pressure of the lubricant which is under a head supply from a tank situated slightly above the load draught of water. Under normal working conditions, oil being lighter than water, there is a slight excess external pressure due to the load draught.

A pump is fitted for charging the system, and an air pipe and drain pipe, so that at all times the shaft runs in an oil bath ; the bearings are either made up of *lignum vitæ* strips or of white metal, which is provided with longitudinal recesses for circulating the oil freely in the bearings.

The system is varied somewhat according to the special case, but the general principle is the same in all.

A somewhat similar arrangement has been tried with success in the "A" bracket bearings of small craft and destroyers, and there is a probable future extension of the system to other and larger vessels. In destroyers and other high-powered vessels the use of white-metal linings for the "A" bracket bearings is common, and only a very small clearance is allowed of about one-thousandth of an inch for each inch of diameter of shaft; under such conditions the fitting of an oil lubrication system is not difficult, and a further extension may be expected in this direction also. In "A" bracket bearings it is necessary to fit glands at each end of the bearing and to lead the supply pipe up to some convenient position inside the vessel immediately above the bracket; this can be easily arranged, and an oil-containing strip (Fig. 168) would be useful at each end inside the packing. Good lubrication of this bearing would save considerable renewals of metal linings.

Safety Stops for Bearing Bolts.—The absolute prevention of the nuts on bearing bolts from slacking back is of the utmost importance, and there are patented arrangements for this purpose. If a bearing is too tightly screwed up it leads to overheating. Any undue slackness allows *hammering*, which gradually increases the slackness by spreading the white metal over the edges of the bearing surface and fills the lubrication grooves, and eventually some damage is done either by overheating from excessive friction and rough surfaces or in a bad case one of the bolts may fracture; this is the usual effect. Good practice requires exactly the right thickness of liner between the halves of the bearing and an efficient and trustworthy stop to prevent slacking back.

A stop suitable for large engines is shown in Fig. 182. The nut *A* is spigoted into a washer ring, *B*, which is prevented from turning by one or two dowel pins, *C*. After the nut is hardened up, a set screw, *D*, is entered into a groove cut in the spigot, and secures the nut from slacking back. A series of countersunk holes should be made, by the point of a drill, in the groove, so that the case-hardened point of the set screw, *D*, should always enter one of them, as shown. The head of the set screw is prevented from turning by passing a socket over it, which is contained in a bracket, *E*, fixed to the body containing the bearing by square-necked studs, nuts, and split pins. As an additional precaution, the bolts are generally fitted with a substantial

split pin, *F*, outside the nut, and with steel washers in any space between it and the nut when hardened up. For convenience in adjustment, the corners of the nuts should be numbered and a radial dial cut into the face of the washer, so that the settings can be recorded and checked when making a fresh adjustment.

Another method used with fast-running engines is shown in Fig. 183. The washer forms part of the nut, and across its edge a number of semicircular grooves, *B*, are cut, generally twelve. Below the face of the washer, and in line with these grooves, holes, *C*, are drilled and tapped, generally ten; and two long studs, *D*, can be fitted, passing

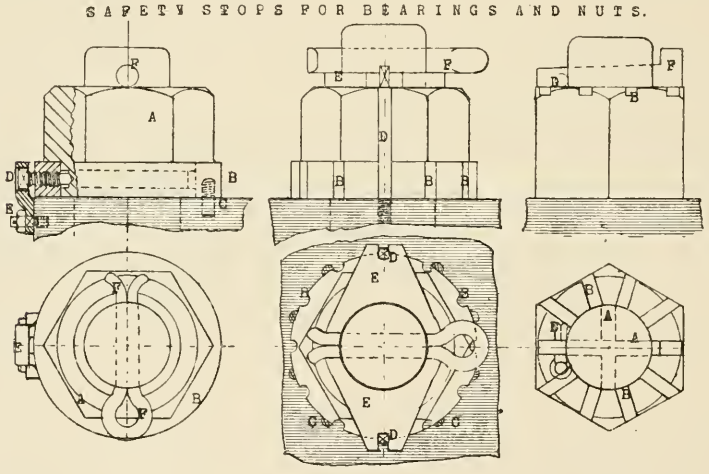


Fig. 182. Fig. 183. Fig. 184.

Safety Stops for Bearing Bolts and Nuts.

through two grooves and into two holes; these studs fix the nut in the required position. The studs are made with squares on them, which are held from turning by a plate, *E*, fitted under the split pin, *F*, and across the end of the bolt. If the screw thread on the bolt is four to the inch (a usual number), and the nut can be set in thirty positions, agreeing with the number of holes given above, then it is possible to make an adjustment of $\frac{1}{120}$-inch on the bearing.

The method shown in Fig. 184 is generally used for piston-rod nuts. A taper key, or cotter, *F*, is fitted in the top end of the bolt and partially sunk into the nut. By cutting two keyways, *A*, in the end of the bolt at right angles, and providing five slots, *B*, in the nut, an adjustment

of one-twentieth of a turn of the nut is obtained; if the number of threads to the inch is four, a longitudinal difference of $\frac{1}{80}$-inch is obtainable. The cotter is secured in place by a split pin, *D*.

Size of Shafting.—If hollow shafting is used, and it is generally, it must be made about 3 per cent greater in diameter for the same strength as a solid shaft, allowing the diameter of the hole to be 55 per cent of the external diameter. The saving in weight is then about 26 per cent on the solid shaft for the same strength. For a certain large engine the diameters of the typical parts of the shafting are as below:—

	inches.
Crank pin	23·25
Crank shaft	21·05
The crank shaft has to take a large bending moment as well as torsional stress.	
Thrust shaft	20·1
This is most difficult to repair, and between the collars it is in addition made $\frac{1}{4}$-inch greater in diameter than at the outer ends.	
Intermediate shaft	19·5
This is most easy to repair and made smallest in diameter, but as it is generally well supported, is nearly as strong as the remainder.	
Stern shaft, inside tube	21·0
This is difficult to repair, almost inaccessible.	
Stern shaft, outside ship	23·0
Frequently of great unsupported length, and therefore subject to a bending moment due to its own weight and also to the working of the ship; also subject to corrosion from sea-water and galvanic action.	

Adopting a factor of safety for torsion of not less than 6, and for tension of about 8, the diameter of the crank shaft is given by the formula—

$$D = F \sqrt[3]{\frac{I.H.P.}{N}}$$

and other diameters are in proportion, where—

D = diameter in inches.

N = number of revolutions of the engine shaft per minute.

F = 4 to $4\frac{1}{2}$, according to the quality of the material.

In the above formula it should be carefully noted that for the same power the diameter of the shaft can be decreased by increasing the rate of revolution of the shaft. This fact is one of the most important items

in the reduction of weight of machinery, and is particularly so in the recent application of geared turbines. In reciprocating engine practice the piston speed is limited, and by decreasing the stroke and correspondingly increasing the rate of revolution the same power can be obtained on a less height and weight of engine. (See also Chapter XVI.)

CHAPTER XIX

THE SLIDE VALVE AND ITS MOTION

Necessity of Slide Valve.—In Chapter I. it has already been explained how, by the addition of heat to water and the consequent formation of steam, the piston can be moved. In the ordinary reciprocating engine the piston is connected by means of the piston rod and connecting rod with a crank arm on a shaft; thus the piston moves only a limited distance each way from its middle position. The total distance which the piston moves is called the stroke, or travel, of the piston, and is equal to twice the *throw of the crank*; the throw of the crank is the distance between the centre of the crank pin and the centre of the shaft on which it is fitted. It is necessary to have an automatic arrangement by which the pressure can be admitted to each side of the piston alternately, and at the same time allow the exhaust steam to escape from, and thus decrease the resistance on, the opposite side of the piston; this is effected by the slide valve. From its peculiar position as the distributor of the steam, the slide valve is a most important element of the ordinary steam engine, as by its means all the processes of admission, cut-off, expansion, release or exhaust, and compression are produced by its proper motion.

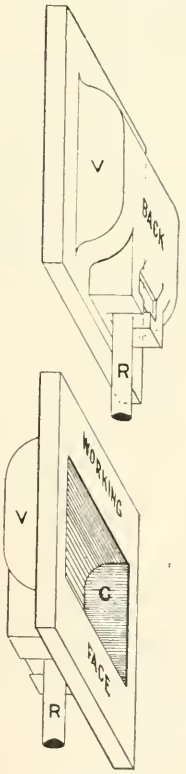


FIG. 185.
Common Slide Valve.

Common Slide Valve.—Fig. 185 shows a perspective view, front and back, of an ordinary D, or locomotive slide valve; Fig. 186 shows the cylinder flat face, on which the valve moves; and Fig. 187 shows a section through the cylinder face with the slide valve in the

middle of its stroke. V is the valve which is moved up and down on the face of the cylinder AB by the slide rod R; the method of obtaining the motion of the valve will be considered later. At A is a steam port, which is in connection with the upper part of the cylinder above the piston; at B is another port, in connection with the lower part of the

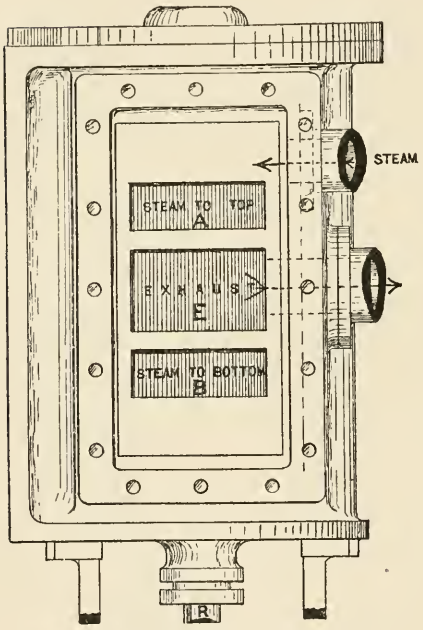


Fig. 186.—Cylinder Face.

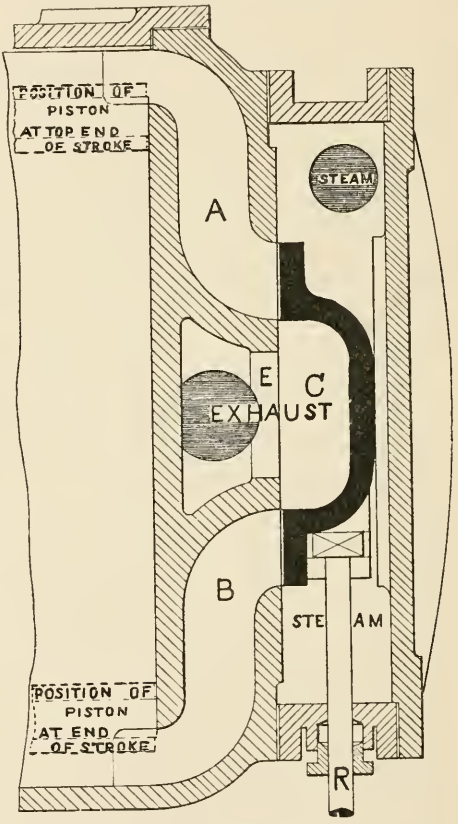


Fig. 187.—Section through Valve and Cylinder Face.

cylinder below the piston; and at E is another port, of about twice the vertical width of the other cylinder ports, A and B, which is connected by a pipe with the exhaust. Opposite to E is a large cavity, C, in the slide valve, known as the exhaust cavity; the travel of the valve is such that C and E are always in open connection.

Steam is admitted to the engine through the regulating valve, and the space outside the valve, away from the cylinder, is filled with

steam under pressure ; the pressure keeps the valve pressed up against the cylinder face. Now, if the valve be moved downwards, the cylinder port A is gradually opened, and steam is admitted above the piston, pressing it downwards ; at the same time the port B is opened and connected through C with the exhaust at E. The difference of pressure above and below the piston produces a motion downwards. Similarly, if the valve is moved upwards, steam is admitted below the piston, the port A is placed in connection with the exhaust, and the difference of pressure produces motion upwards ; and so on, alternately, in accordance with the distribution of pressure by the valve.

The operations performed by the valve are considered later, when the means of producing its motion have been more clearly explained.

Production of Motion of the Valve.—The most usual method of moving the valve is from an eccentric fitted on the shaft. The piston reciprocates and produces a revolving motion of the shaft through the medium of the crank. If another crank is fitted on the shaft, it can be made to work a reciprocating piece, such as the slide valve. This is done in ordinary practice ; but an eccentric is substituted for the crank, as it is not convenient to cut the shaft for the purpose.

In Fig. 188 an eccentric is shown keyed to the shaft, and embraced by the eccentric strap ; the upper half of the strap is secured to the eccentric rod, which at its top end R is connected with the slide rod, either directly or through the link. The eccentric revolves with the shaft, inside the strap, and produces a reciprocating motion of the eccentric rod and slide rod.

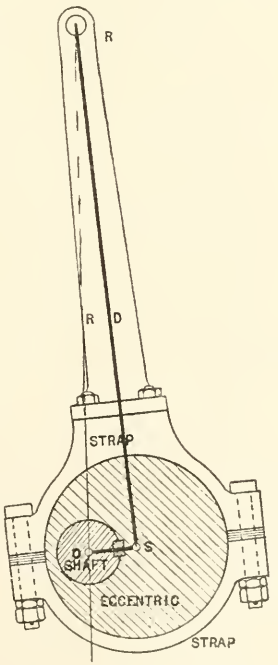


FIG. 188.—Eccentric and Rod.

If the crank pin of an ordinary crank shaft be enlarged sufficiently to include the shaft within it, the pin becomes an eccentric. The eccentric is therefore a crank OS fitted in a special way. The distance between the centre line of the shaft and the centre line of the eccentric OS is equal to the throw of the eccentric or virtual crank, and the travel of the valve becomes equal to

twice the length of the virtual crank, or *radius of eccentricity*, as it is frequently called.

Definition.—The *radius of eccentricity* is the length of the virtual crank, and therefore *one-half the travel* or stroke of the slide valve; it is the distance between the centres of the shaft and the eccentric.

Normal Slide Valve.—The early slide valves were made without lap—that is to say, when the valve was in the middle of its stroke, any motion of the valve opened the cylinder to steam, either above or below the piston. In Fig. 186 a normal slide valve is shown in the middle of its stroke, and with no part of the valve overlapping the cylinder ports on either the steam or exhaust edges. For working the normal slide valve the eccentric radius is fixed at right angles to the crank, so that when the piston is at either the top or bottom of its stroke the valve is near the middle position, and just covers both the cylinder ports.

In Fig. 194 the crank is represented by *OP*, and the eccentric radius by *OS*; in the first position a movement of the crank in a clockwise direction pushes the valve upwards, admitting steam below the piston, and opening the other port to exhaust; therefore the piston will promote rotation in a clockwise direction. The cylinder is open to steam from below during the upward stroke of the piston, and the upper side of the piston is open to exhaust. Both ports are closed near the top of the stroke, and the crank continuing its rotation, from its own momentum or some external cause, steam is next admitted above the piston, and the exhaust is opened to the bottom of the cylinder. The motion produced is therefore in a clockwise direction.

If an eccentric is fixed to the opposite side of the crank, as shown by the dotted lines *OS'*, a movement of the crank in a clockwise direction pulls the valve downwards, admitting steam above the piston, which tends to return the latter to its original position. But if the crank is moved anti-clockwise, the motion of the valve is such that the steam is admitted and exhaust opened to assist motion in the same direction.

From the foregoing it may be concluded *that the eccentric leads the crank in the direction of rotation.* This is true, however, only when the steam enters the cylinder from the outer edges of the valve (as in Figs. 185 and 189), and generally for all flat valves. The cylindrical valve shown in Fig. 204 is an example of an exception.

Lap.—The normal valve admits steam for the whole of the stroke

of the piston. It is economical, however, to arrange for the steam to be used expansively by cutting off the supply at some point before the end of the stroke. This is effected by making the valve overlap the steam ports of the cylinder, as in Fig. 189. If lap is fitted as in Fig. 189, and the eccentric radius is set at right angles to the crank, it is evident that the crank must be moved a certain distance before steam is admitted to the cylinder. Although the cut-off now takes place earlier, the steam admission takes place later, and the amount of opening to steam has been decreased.

To obtain the same maximum opening to steam, it is necessary to increase the travel of the valve, that is, to increase *OS*; but after this is effected the steam still enters the cylinder too late.

By advancing the position of the eccentric radius relative to the crank until the valve begins to open at the beginning of the stroke of the piston, and following up the operations of the valve with respect to the piston, it will be noticed that all the operations are performed as required, viz. admission, cut-off, expansion, and release.

Definition.—The lap of the valve is the amount the valve overlaps the cylinder steam port when it is in the middle of its stroke; the amount of overlap on the steam edge is called the *steam lap*, and the amount it overlaps the exhaust edge is called the *exhaust lap*.

The addition of steam lap to the early locomotive engines produced an economy of fuel of 25 to 50 per cent.

Exhaust lap is very small in compound expansion engines, and is frequently *negative* (Fig. 189).

Lead. *Definition.*—The amount of opening of the port to steam when the piston commences its stroke is called the lead of the slide valve.

When steam is admitted at the beginning of the stroke, it takes an appreciable time to fill the cylinder, as the admission is gradual; therefore the full pressure does not act on the piston at once. At the end of the stroke the reversal of direction of motion of the piston produces great shock, unless some means are taken to cushion it just previous to the reversal. By advancing the eccentric a small amount (Fig. 190), the slide valve is given sufficient lead to admit steam before the end of the stroke. This forms an elastic cushion, on which the piston is brought gently to rest, and at the same time assists the reversal of its motion. The increased advance of the eccentric used to obtain the lead makes all the operations performed by the valve earlier.

Angle of Advance of the Eccentric. — The total angle the

eccentric is advanced for the necessary lap and lead is called the angular advance.

The angle varies for different types of valves and engines, but is generally about 40°. By reference to Fig. 190, a little consideration makes it evident that when the eccentric radius *OS* is at right angles to the line of travel of the valve, the valve is not quite in the middle of its stroke, but rather closer to the shaft ; this is due to the obliquity of the eccentric rod. The length of the eccentric rod is very long compared to that of the eccentric radius, and consequently the obliquity is very small ;* and by neglecting obliquity the following construction gives the angle of advance. In Fig. 190, *OP* represents the position of the crank. *OL* is made equal to the lap, and *LD* is made equal to the lead. Make *OS* equal to the radius of eccentricity or half-travel of the valve, and describe a circle with *OS* as radius ; draw *DS* perpendicular to *OD* and cutting the circle at *S* ; then *OS* is the position of the eccentric radius or arm, and the angle *KOS* is the angle of advance.

Definition.—The angle of advance may also be defined as the angle at which the eccentric radius stands in advance of the position that would bring the valve to its mid-stroke, when the crank is on the dead point (*i.e.* in line with *OR*).

Width of Exhaust Cavity.—So far, only the steam edges of the valve have been considered. These govern the admission and cut-off of the steam, while its compression and release are governed by the exhaust edges. It is arranged that the exhaust is as free and unrestricted as possible, but it is necessary that the exhaust should be closed before admission commences. The decreased pressure, obtained by the expansion of the steam in the cylinder, produces a large increase in volume, which has to pass through the port in equal time, *i.e.* at each stroke ; consequently the width of the cylinder ports, through which the exhaust has also to pass, is about 50 per cent greater than the maximum amount that the valve uncovers of the ports at the steam edges ; but the full width of each port is uncovered by the exhaust edge of the valve.

The width of the exhaust cavity in the valve must be such as to comply with the above conditions, and also to suit the width of the exhaust port fitted in the cylinder false face.

Operations of the Slide Valve.—In Fig. 191 a series of positions of the crank, the piston, and the slide valve are shown relative to each other. For clearness the eccentric radius is shown as a crank arm turning with the main shaft. *OP* represents the crank, *OS* represents

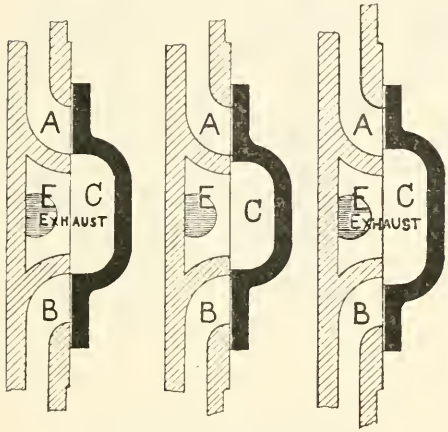
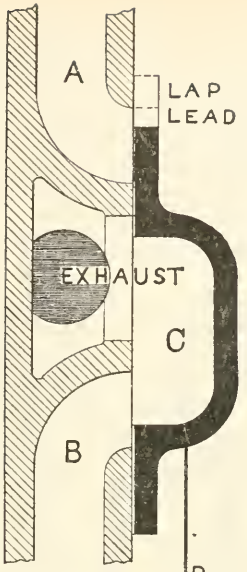


FIG. 189.—Lap of Slide Valve.



$OL = O_1L_1$ = Steam LAP.
$LD = L_1D_1$ = Steam LEAD
Angle KOS = Angle of Advance of Eccentric due to LAP + LEAD

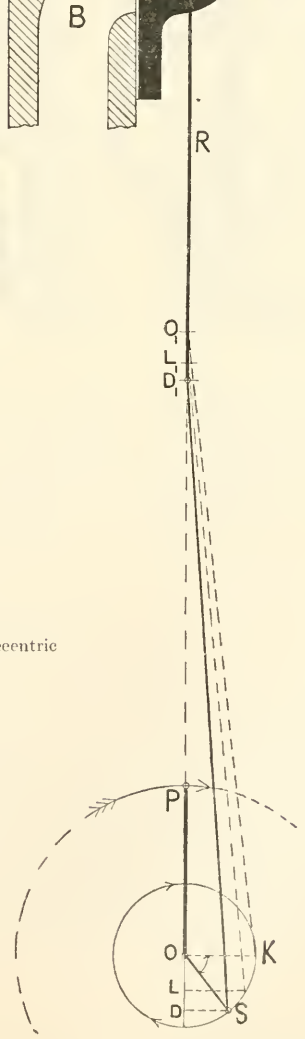


FIG. 190.—Angular Advance of the Eccentric.

the eccentric arm, and the excess of the angle *POS* over a right angle is the angle of advance.

Admission.—When moving in the direction of the arrow, the piston and crank are moving upwards. Just before the end of the upward stroke of the piston, the valve moving downwards uncovers

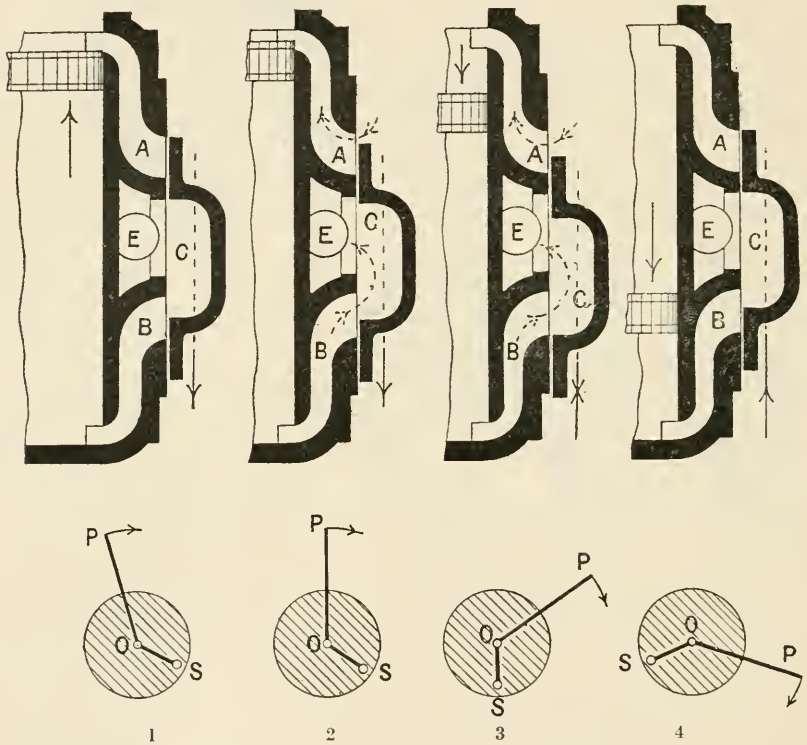


Fig. 191.—Operations of the Slide Valve.

1. Admission.
2. Lead, Down-stroke.
3. Maximum opening to steam.
4. Cut-off, Down-stroke.

the upper cylinder port and admits steam above the piston; the instant at which the port begins to open is called the instant of admission. Owing to the lead, the piston has not quite reached the top of its stroke when admission begins.

Cut-Off.—When the crank reaches the position shown in figure (4), the piston has travelled some distance downwards while the valve has travelled downwards, and, returning upwards, cuts off the

admission of steam ; the steam expands as the piston continues its stroke until release takes place.

Release.—The motion of the valve upwards next uncovers the upper cylinder port connecting it with the exhaust, and the instant at which the port begins to open is called the instant of release ; the

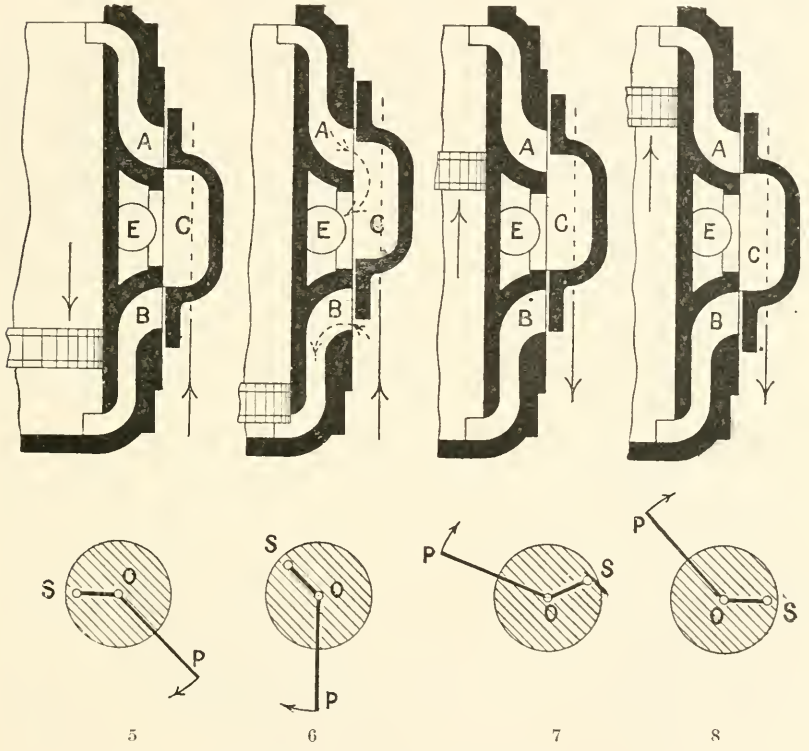


Fig. 192.—Operations of the Slide Valve.

5. Release, Down-stroke.
6. Lead, Up-stroke.
7. Cut-off, Up-stroke.
8. Compression, Up-stroke.

steam escapes from the cylinder into the exhaust cavity through the upper cylinder port.

If there be zero exhaust lap, the instants of release and compression (which latter is next described) are coincident ; if there be positive exhaust lap, compression begins before release ; but if there be *negative* exhaust lap, which is more usual, release begins before compression.

Compression.—At this instant no steam can pass into the cylinder,

as both ports are closed, and no exhaust can take place for the same reason. The bottom port, *B*, has just closed connection with the exhaust; the steam remaining below the piston is imprisoned, and compression begins. If the link, at this point of the stroke of the piston, is drawn over towards its position for astern working, it will be noticed that compression would have begun earlier; *i.e.* by "linking-up" or "shortening the link," compression begins earlier.

Readmission.—At a point similarly situated to that of admission, but on the down-stroke, compression begins below the piston just previous to the instant of readmission, *i.e.* admission for the up-stroke. At the instant of admission for the up-stroke the crank has advanced to nearly the bottom dead centre, the steam above the piston has been released by the valve opening to exhaust through the cylinder port *A*, while the steam below the piston has been compressed by the valve covering the exhaust through *B*, and *B* is just at the instant of admitting steam.

On the up-stroke, the process of cut-off, release, compression, and readmission for the down-stroke are repeated; and so on in continuous rotation.

Eccentric, Strap, and Rod.—The details of the construction of an eccentric and strap for a large engine are shown in Fig. 193. The eccentric, or sheave, is sometimes made in one piece and slipped over the end of the shaft and keyed to it; but this is only possible in a few instances. Generally it is made in two parts, *A* and *B*, tightly embracing the shaft, *C*, and secured together by two bolts, *D*. The two parts are filleted at the butting faces, which are machined up to true surfaces. The smaller part, *B*, is generally made of forged steel, the greater strength of which allows the outer part of the sheave to be brought more closely to the centre of the shaft, and thus reduces the diameter of the eccentric. The larger part, *A*, is generally made of cast iron, lightened by holes, and keyed to the shaft by one or more keys. The key should be carefully fitted, and made a driving fit on its sides and only a moderate fit radially, so that the part, *A*, as well as *B*, should be in firm contact with the shaft throughout its circumference. Two set screws, *E*, with check nuts, are used for temporarily fixing the eccentric in position on the shaft when setting the slide valve, before the actual keyway is cut. They are also used to prevent the sheave creeping along the shaft, and, if the fit is faulty, to take up any slackness between the shaft and eccentric.

The right-hand part of Fig. 193 shows a pair of eccentrics and straps:

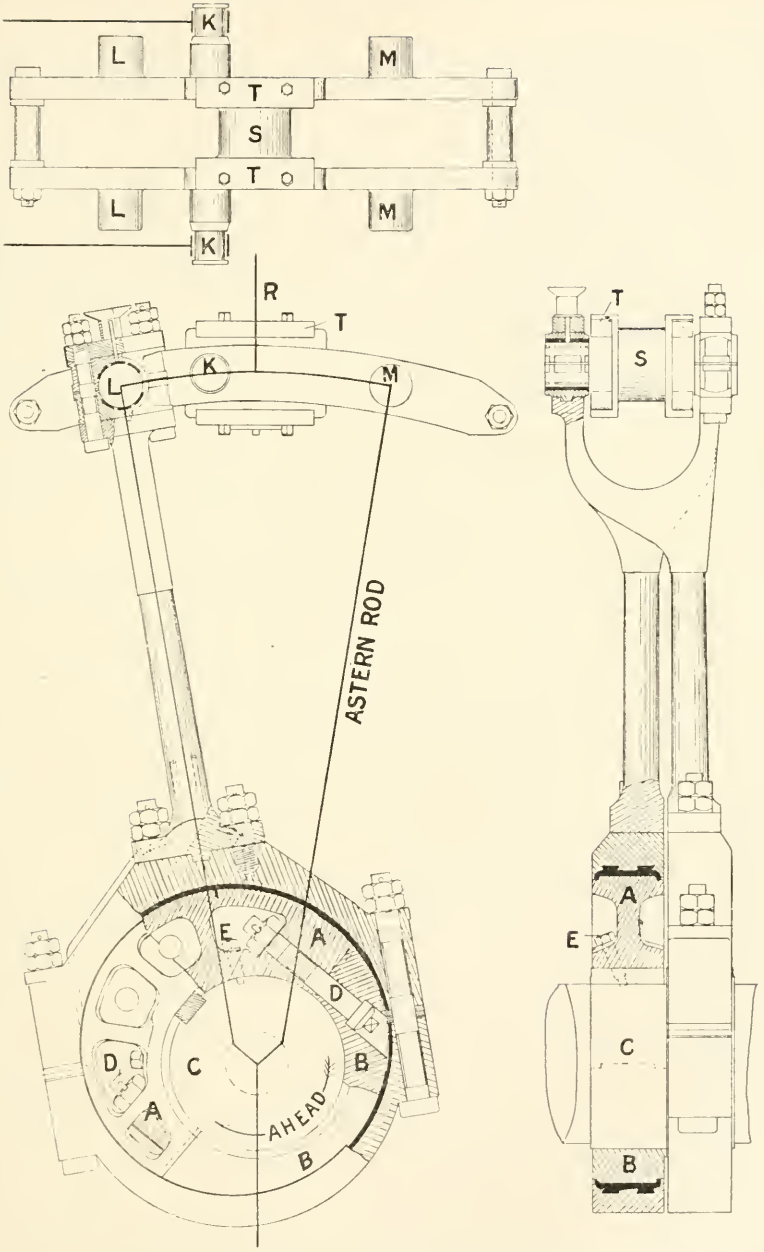


Fig. 193.—Eccentric, Strap, and Rod, with Link and Link Block.

the wide one, which is in section, is used for ahead working ; and the other, shown in side elevation, is used for astern working. The sheaves fit over an enlarged part of the shaft, so that the cutting of the necessary keyways does not weaken it.

The *eccentric strap* is made in two parts, generally of steel, but sometimes of gun metal, and in either case is lined with white metal, which forms the rubbing surface with the sheave. The parts are made with an inwardly projecting flange on each side, to prevent the oil being thrown out too easily by centrifugal action, and these flanges also maintain alignment between the eccentric and slide rod. Gun-metal liners are fitted between the butting ends of the upper and lower parts of the strap : the thicker liners form distance pieces, and do not touch the sheave ; the thin liners are used for correcting adjustment. When the engine is at work the straps tend to contract at the lips, especially if the bolts are not sufficiently tightened up on the liners, and to produce sufficient stiffness the straps are sometimes made with ribs.

The *eccentric rod* is made of steel, with a foot, and a spigot below the foot, which fits into a recess in the top part of the strap, and maintains alignment. Two studs, generally with collars and nuts, are used to secure the eccentric rod to the shaft.

The *bolts* securing together the two parts of the sheave and the two parts of the strap are made of steel, with the usual stretching lengths, nuts and washers, and split pins, and a socket key under the head of the bolt, to prevent its turning in the hole. Cotters were sometimes used instead of nuts, but they are not so satisfactory, principally because the butting surface is not large enough.

Setting the Slide Valve.—The eccentric is first secured temporarily in place at the supposed correct angle of advance, and where reversing motion is fitted the astern eccentric and its connections are also temporarily secured in their proper places. The crank is then placed first on the top centre and then on the bottom to verify the setting of the slide valve. When the valve is correctly set the top and bottom leads *with the link in full gear* should be equal, and for " ahead " full gear the position of the valve on the valve rod is altered if necessary to obtain equal top and bottom lead. Next, the proper *amount* of lead is obtained by altering the angle of advance of the " ahead " eccentric. For " astern " full gear, the length of the " astern " eccentric rod is adjusted to give equal top and bottom lead, and afterwards if necessary the angle of advance of the " astern " eccentric is altered to produce the required lead.

After making the above adjustments, the position of the valve on its rod may be altered to give not more than 30-thousands of an inch more lead on the bottom than the top in a large vertical engine, to allow for subsequent wear down the link motion and eccentric motion bearings. All the leads should then be checked over and the position of the piston in its strokes, up and down, carefully noted with respect to the position of the valve at the instants of admission, cut-off, release, and compression. Any error in these readings points to inaccuracy of lead and particularly to an inexact position of the crank at the time of measurement. If the leads particularly are correct the other readings must be correct also, and the keyways may be cut and the eccentrics permanently secured in position on the shaft.

A calliper gauge is generally made to suit marks made correspondingly on the fixed guide and on the moving slide rod for the full link positions, so that the leads can be verified at any time without removing the slide box covers and exposing the valves.

In sea-going ships, inequality in the lengths of the "ahead" and "astern" eccentric rods is a common defect in the setting. With equal length and a correct position of the valve on its rod it is possible, with the usually well-designed link motion, to work with a very early cut-off and consequent economy. For any position of the link and at any power the mean pressures of the upward and downward strokes, as shown by the indicator diagrams, should not differ by more than 10 per cent.

Reversing the Engines.—The direction of the rotation of the engine shaft may be reversed by fitting two eccentrics at the relative positions *OS* for "ahead" and *OS'* for "astern" working (Fig. 194), and then connecting the top ends of the eccentric rods by a link *RR'* with which the valve rod is connected by a link block. The position of the link relative to the block is governed by bridle rods connected with the reversing gear or lever (see Fig. 195). The act of moving the link by the reversing lever alters the position of the valve on the cylinder face, as shown in the three figures. In the "ahead" position, shown in the left-hand figure, the movement of the valve is almost entirely governed by the "ahead" eccentric rod *RS*; and the crank *OP* having passed the dead centre and begun the up-stroke, the valve allows the steam to enter below the piston and thus continue the rotation of the crank in the "ahead" direction.

In the "astern" position shown in the right-hand figure the movement of the valve is almost entirely governed by the "astern" rod

$R'S'$ which is pushing the valve upwards so that release is about to begin, and when the crank reaches the bottom position the lead of the valve allows the steam to enter below the piston and continue its motion in the upward direction necessary for " astern " motion.

In the " mid-link " position, shown in the middle figure, the movement of the valve is governed by both " ahead " and " astern " rods,

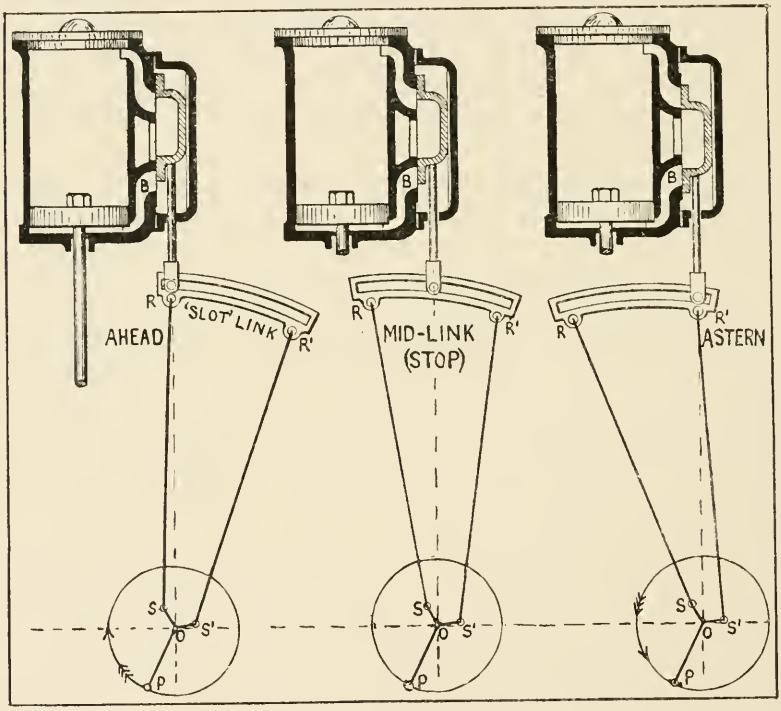


Fig. 194.—Reversing with Two Eccentrics.

but the total movement is slight and insufficient to produce continuous rotation of the shaft in either direction.

The Link.—There are three varieties of link, and room must be given in all of them for the oscillation of the block to which the slide rod is attached. For the purposes of reversing only, the link could in all cases be made straight; but in practice it is generally curved, so that it can be readily set in any intermediate position.

For a perfect linking-up the curvature should be parabolic; but for practical purposes it is part of a circle, whose radius is equal to

the length of the eccentric rod measured from the centre of the link, *i.e.* the centre of the pin joint in the end of the slide rod.

The *slot link* is still used for small engines, and was also frequently fitted in horizontal engines. The connections of the eccentric rods are outside the slot; and in a vertical engine they are below it, and nearer the eccentrics.

The *double-bar link* is the most frequently used for marine engines. It is shown in Fig. 193. Two curved bars are joined together at the ends by bolts, fitted or made with distance pieces. The block *T* slides along the bars between these distance pieces. Each of the two pairs of pins, *L* and *M*, fitted on the outer side of the bars is connected with one of the eccentric rods, which are forked at the top end, for the purpose of fitting twin bearings. These forks are right- and left-handed, rising from the eccentrics, which are fitted side by side on the crank shaft. The *bridle rods* are attached to the central (or other) pins *K* at their outer ends inside the collar shown, and by their means the link can be moved to or fro to bring the required eccentric rod in line with the slide rod *R*, through the medium of the link block *T*.

The **link block** travels inside the double bars of the link, and is generally provided with a white-metal rubbing surface where it is in contact with the upper and lower edges of the bars. Its central part is provided with a pin, *S*, which is attached to a bearing in the end of the slide rod. The link block is adjustable, and the white-metal lining is generally contained in brasses, secured to the steel block by screws, as shown.

White metal is now generally used for all the working surfaces of the bearings, for the pin joints, *L* and *M*, of the link and weigh-shaft arms *K* and their connections; but the bearings of the weigh shaft itself are of gun metal.

The *solid-bar link* is sometimes used. The link is one solid bar, and can slide between a pair of sector blocks, which are fitted to, and can oscillate in, the end of the valve rod. This arrangement necessitates fitting the connection with the eccentric rods at the extremities of the link, and therefore neither eccentric rod can be brought even approximately in line with the slide rod. The link is consequently "shortened" or "linked up," even when in "full gear," and to obtain the necessary extreme travel of the valve, a greater throw and larger eccentrics must be fitted than with the two types previously described. The larger eccentrics tend to increase the frictional resistance; and

although the solid-bar link appears to work fairly well in practice it cannot be made so efficient as the double-bar link.

Linking-up.—If the link block is set at a point R midway between the two points L and M, the motion imparted to the valve rod is partially neutralised, and any steam admitted tends to keep the piston in the middle of its stroke or thereabouts, and does not allow of a complete revolution of the crank.

If the link be set at some point, say, halfway between L and the mid-position, the motion imparted to the valve is derived from both ahead and astern eccentrics, but principally from L, and the distance travelled by the valve is considerably less. If the linking-up, either on the ahead or astern side, be not too great, and all the necessary functions of the valve be performed, motion is imparted by the pressure through the openings as before, and the direction of rotation depends on which of the rods, ahead or astern, predominates on the travel of the valve. (See also Fig. 194.)

When linked up, all the operations of the valve are earlier—namely, admission, cut-off (producing greater expansion), release, and compression.

Reversing Gear.—There are three principal methods of reversing the motion of the engine in everyday use :

1. Changing over the entry and exit of the steam and exhaust, by using a *differential reversing valve.* This system is used for steering and other engines, and is explained in Chapter XXIX.

2. *Valve gear,* either with one or no eccentric, as explained later for Joy's arrangement ; the motion is then obtained from the movement of the piston or connecting rod. (See Fig. 210.)

3. Stephenson's **link motion** (Fig. 193).

The most usual method is that adopted by Stephenson, which, although modified in detail, is essentially the same in principle as when first fitted in the *Rocket.* Normal slide valves were used with the original link motion, with practically no lap or lead, and consequently expansion was almost non-existent.

Reversing Gear, All-round Type (Fig. 195).—The link motion is shown in the ahead position, and the crank is at the top of its stroke ; the slide valve is of the cylindrical type, taking steam at its mid-length.

The movement of the link to and fro, as necessary for altering the direction of rotation of the engine, is generally obtained by the **all-round** or continuous-motion reversing gear, as shown in Figs. 195 and 196. The notation used in these two figures is similar, but the

weigh shaft is differently arranged. In Fig. 196 the crank is at the bottom of its stroke, and the link is central. The gear may be worked either by hand or by steam, and the reversing engine need only be capable of moving in one direction; but frequently a differential valve is fitted with a two-cylinder engine, and the motion is then reversible.

In Fig. 195 the shaft AC is revolved either by the hand-wheel A or by the reversing engine acting on the crank B. A worm C is attached to the shaft, and operates a worm wheel D on which a pin E is fixed. A connecting rod EF works an arm FG keyed to the weigh shaft G. Another arm HG is also keyed to the weigh shaft, with the end H connected by a rod (or rods) HK with the link LM at K. R is the centre of the crank shaft; RL and RM are the eccentric rods. According to the direction of motion of the reversing engine, E is made to revolve in a circle, and moves F from right to left or left to right. In this way the link is dragged to and fro, to bring the rod RM or RL in line with the slide rod, and thus give the desired position for either ahead or astern rotation of the main engine.

This motion is very quick in its action, and requires very careful balancing of the reciprocating parts of the engine, so that it can be stopped in any required position of the link. A double-cylinder reversing engine with a differential reversing motion is preferable for use with all types of main engines. With a continuous-motion gear the distance through which the links are moved is governed by the travel of the pin E of the wheel D, and there is no chance of accident from jambing in any position. When the movement of the link is derived from a direct motion by a piston or screw gearing, any slight derangement of the automatic cutting-off of the steam to the reversing engine may result in the link motion being jambed in an extreme position, and produce a breakdown.

The "all-round" arrangement is very convenient when waiting orders or raising steam; the reversing engine can be kept continuously in motion, and thus allows the steam to uniformly warm and free the main engines from water. The slight movement of the main engines when the reversing gear is in motion is not objectionable, and frequently produces a slight vacuum in the condenser if the air-pump is attached to the main engines. A convenient and expeditious means of warming the main engines is almost a necessity with water-tube boilers, where the time required for raising steam is governed by the time required for properly warming the engines.

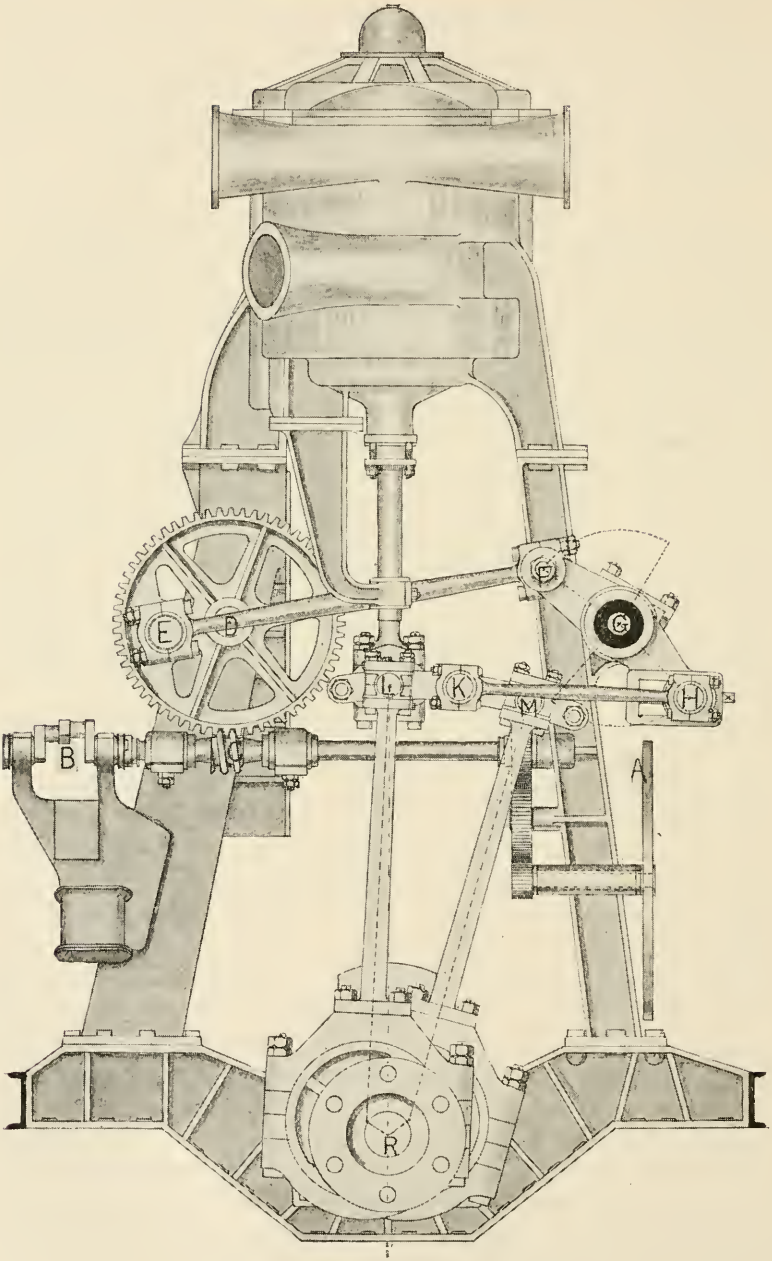


Fig. 195.—All-round Reversing Gear.

In Fig. 195 the general arrangement of the cylinder and supports is the same as in Fig. 142. The extreme positions of the reversing arm and lever, the centre lines of the eccentric rods, and the eccentrics, are dotted in. The link block is behind *L*, and is only partly visible ; just above the block a circular guide is fitted for the slide rod in the end of the bracket shown.

In both Figs. 195 and 196 the eccentric rods are **open**—that is to say, when the eccentric radii point towards the link the rods are not *crossed*. In Fig. 195 the eccentric *does not lead* but follows the crank in the direction of rotation, because the steam port in the cylindrical

References to Fig. 195

A, Hand-reversing wheel.
B, Cranks of reversing engine.
C, Worm on reversing shaft, gearing into worm wheel.
D, Centre of worm wheel.
E, Pin on worm wheel, making *DE* a crank.
EF, Connecting rod to reversing lever.
F, Pin on reversing lever.
G, Centre of weigh shaft.
GH, Reversing arm keyed to weigh shaft.
H, Twin pins of the *separate link*, which is incorporated in reversing arm.
HK, Suspension or bridle rods.
K, Pins on link to which the bridle rods are attached.
L, Pins on link to which **ahead** eccentric rod is attached.
M, Pins on link to which **astern** eccentric rod is attached.
LKM, **Double-bar link** (Fig. 193).
R, Centre of crank shaft.

slide valve fitted to the cylinder admits steam from between its inner edges, and not at the outer ends.

Separate Link.—In the arm of the weigh shaft *G*, which is connected with the drag links, or bridle or suspension rods, as shown in Figs. 195 to 197, a slot is fitted with an adjusting screw and block, *H* ; the block forms the connection with the bridle rods through a pin joint. When the main link is in the proper position for ahead working, it is approximately in line with the slot in the arm, and a movement of the block, which is effected through the screw, draws the link over to any desired position. If the main link is in the astern

position, the slot in the arm (or separate link) is at right angles to the line of movement of the main link, and any movement of the separate link does not affect the relative position of the main link

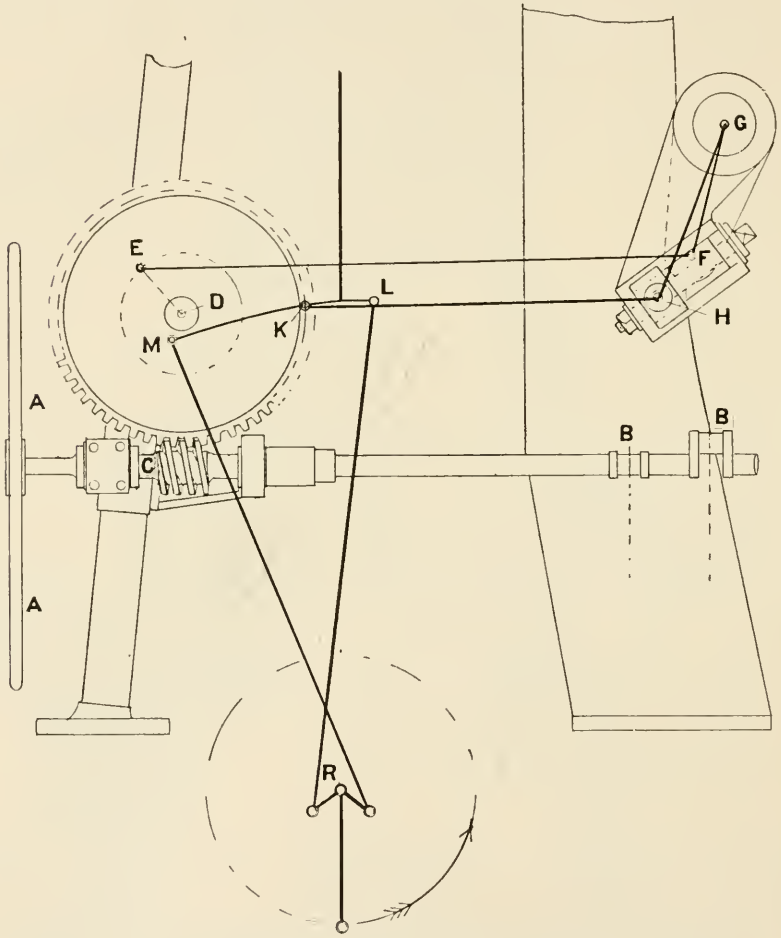


Fig. 196.—All-round Reversing Gear.

and its block, to which the slide rod is attached. Consequently the main engines can always be readily reversed, independently of the separate linking-up.

This means of subdivision is fitted in nearly all high-powered engines, and can be used at any time with safety and economy.

Although an equal division of power may generally be considered as desirable, it does not follow that it gives the best economy—that can only be obtained with a due regard to the range of temperatures in the cylinders in conjunction with the actual coal consumption in practice.

At low powers, it is preferable to link up as far as possible consistently with smooth working, and at the same time to reduce the working pressure to the lowest safe limit for reversing and general purposes ; this reduces the range of temperatures in the cylinders, and the loss consequent on cylinder condensation.

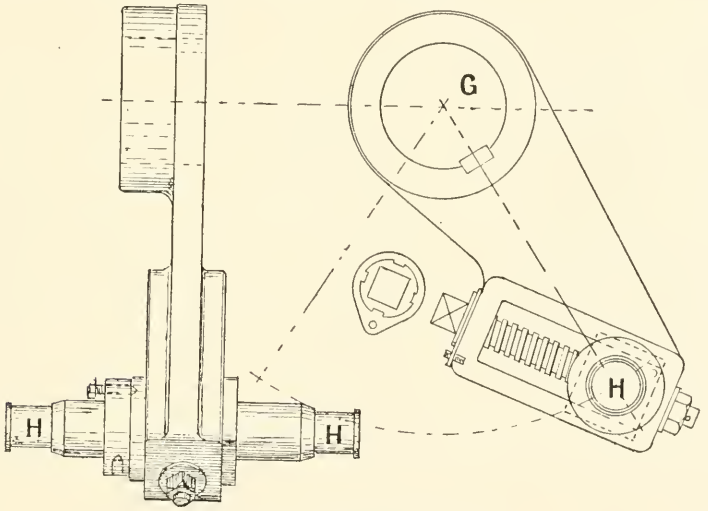


FIG. 197.—Independent Link.

Independent Alteration of Cut-Off.—The weigh shaft operates each link in exactly the same way, and any linking-up so obtained tends to alter the cut-off equally in each cylinder. For lower powers this is not always satisfactory, and the effect of altering the cut-off in each cylinder independently will now be considered. In Fig. 198 the shaded portions of the figures represent the volume occupied by the steam in each cylinder in turn at the instant of cut-off. It may be considered that the same weight of steam occupies each cylinder in turn, for each piston makes the same number of strokes per minute.

1. *Cut-Off earlier in H.P. Cylinder.*—By cutting off earlier in the H.P. cylinder, a lesser volume of steam is admitted at each stroke, the expansion is greater, and consequently the mean pressure on the

H.P. piston is less. The pressure admitted to the M.P. and L.P. cylinders is also less, and their mean pressures are less, and consequently there is a general reduction in the I.H.P. and the number of revolutions of the engine. For lower powers, this method is generally used to obtain economy by using increased expansion.

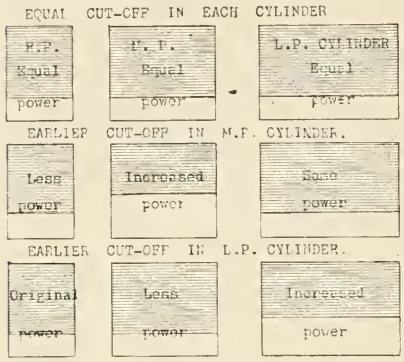


FIG. 198.—Effect of Alteration of Cut-off.

2. *Cut-Off earlier in the M.P. Cylinder.*—The exhaust steam from the H.P. cylinder occupies less space in the M.P. cylinder, and consequently the back pressure in the H.P. cylinder is increased, and reduces the work done in that cylinder. The same weight of steam is admitted to the M.P. cylinder as before, but as it occupies less space, the pressure is greater. The exhaust from the M.P. cylinder is not affected, as it fills the same space as before, in the L.P. cylinder; and consequently, as the driving pressure is higher and the back pressure is the same, the mean pressure is greater, and more work is done in the M.P. cylinder.

3. *Cut-Off earlier in the L.P. Cylinder.*—On the same reasoning as for the M.P. cylinder, the work is decreased in the M.P. cylinder and increased in the L.P. cylinder.

It is thus seen that by linking up for any one cylinder or cylinders independently, a certain subdivision of power can be obtained, and for this purpose the separate, or independent, link is used.

Double-ported Slide Valve.—The opening to steam through the cylinder port is governed by the movement of the slide valve, and the comparatively long stroke or travel of a single-ported slide valve requires a large radius of eccentricity and a correspondingly large eccentric, with a very large frictional loss. To reduce the travel and the frictional loss, the single-ported valve has been replaced by double- or treble-ported valves, which give the same opening for about one-half or one-third the travel or stroke.

In the series of figures 199 the derivation of the double-ported valve from a single-ported valve is shown in stages. First, the single-ported valve is shown, with its two cylinder ports for steam and one for exhaust. In the next stage the original valve is partly covered

by a diaphragm, which contains the steam pressure about the valve ; outside this diaphragm a large valve is fitted, working over cylinder ports of similar width to the original ports fitted for the small valve. The three parts—the inner valve, the diaphragm, and the outer valve—are incorporated in one piece, and they travel together over the cylinder face, within a slide jacket or cover, which contains the steam pressure about the outer valve. The necessary connection of the exhaust space outside the diaphragm, with the exhaust space within the inner valve and the cylinder exhaust port, is made by cutting away the back of the inner valve and part of the diaphragm, as shown in the third stage, and enclosing the steam ports of the valve.

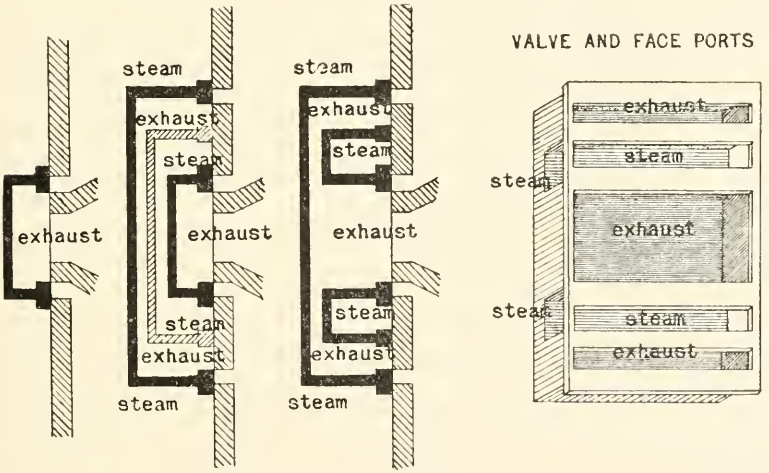


FIG. 199.—Principle of Multi-ported Slide Valves.

The fourth stage shows the front face of a double-ported valve with its steam and exhaust ports and their connections, with the steam outside the valve and cylinder exhaust port, respectively.

All the operations of the double- or treble-ported valve are precisely similar to those performed by the single-ported variety.

The double-ported valve (and treble-ported valve proportionately) takes up more room and increases the length of engine ; there is greater wire-drawing, and consequent loss of velocity of the steam, from frictional resistance at the edges of the valve and cylinder ports, as their length is increased by the employment of multi-ported valves ; but the reduction of travel with the decreased size of eccentric necessary, more than counterbalances these disadvantages.

In Fig. 200 a type of double-ported slide valve, as fitted to the L.P. cylinder of a large engine, is shown at about the middle of its stroke. The valve rod runs through the vertical middle of the valve, and is allowed a very slight play between the collar fitted on a cone at the lower part of the rod and the check nuts on top of the valve; this amount should not be sufficient to allow "hammering," but it allows a movement of the valve towards the cylinder face, and the hole through the valve is therefore elongated in the required direction.

At the lower part of the valve chest a large opening is made for a gland and stuffing box (not shown) for the rod. Stays are fitted to some of the flat surfaces in the cylinder ports, and some of those at the top end also form the studs for securing the cylinder cover. The cover for the slide-valve chest or jacket is provided with lugs and holes, *P*, for supporting the weight of the cover when removed.

Relief or Equilibrium Ring.—A flat valve, whether single or multi-ported, is kept pressed up against the cylinder face by the steam pressure acting on the back of the valve, away from its face. When the pressure is great and the area of the valve is large, there is a great waste of work in overcoming the friction thus set up between the rubbing surfaces, and therefore means are provided to relieve the pressure from the back of the valve.

The back, *AA*, of the valve is made flat over as great an area as possible, with its surface parallel to that of the cylinder face; against this back face a packing piece, *C*, called the relief ring, and generally circular, attached to the slide jacket or cover, is pressed by small spiral springs and rods, *B*; the valve reciprocates between the cylinder face and the flange, *CC*, of the relief ring. The space enclosed within the relief ring is connected at *D* with the exhaust side of the valve through the receiver pipe, or, in the case of the L.P. valve, with the condenser. The pressure of the steam on the back of the valve is thus excluded from the area included by the relief ring, and reduces the total pressure on the cylinder face to a corresponding extent.

The simple method of connecting the space contained within the relief ring with the exhaust cavity of the valve is now found to be efficient in practice, but leakage is difficult to detect; the efficiency of the relief afforded can only be tested when the engine is at work. In the usual arrangement, a cock is fitted for detecting leakage by connecting it with a gauge and comparing the reading with the receiver or condenser gauge.

The relief ring is kept steam-tight at its junction with the slide

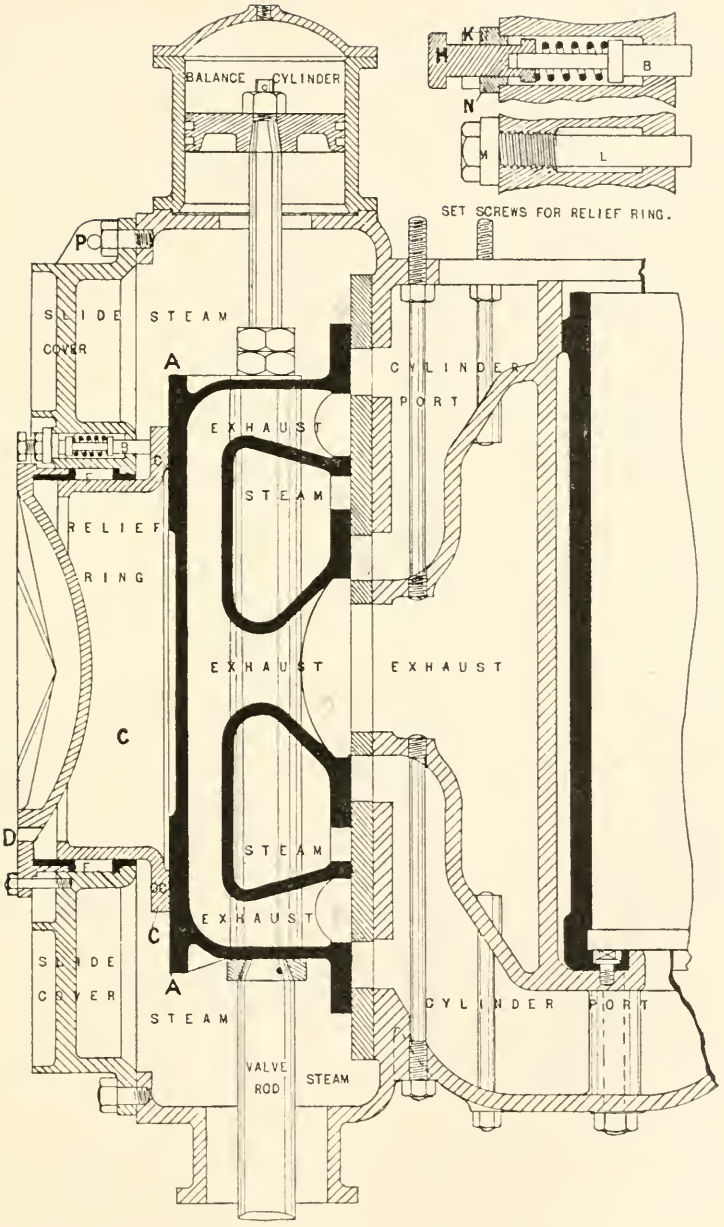


Fig. 200.—Double-ported Slide Valve.

cover by fitting a stuffing box and gland F, packed with asbestos or vulcanite rings, and in some cases Ramsbottom rings are used. In many of the older engines a bellows ring was used ; but with modern high pressures they were found to be very uncertain in their action under varying temperatures, and have necessitated large repairs or alterations.

To prevent the valve being forced away from the cylinder face by a momentary pressure, such as excessive cushioning of the piston, set screws (enlarged views of which are also shown in Fig. 200) and springs are fitted at the back of the packing ring to keep it pressed lightly against the back of the valve. The pressure exerted by the springs is very small, and the set screws, after being screwed up moderately tightly, are slacked back a certain amount, generally from $\frac{1}{32}$ to $\frac{1}{16}$ inch. In the event of the springs breaking, they can be set up more closely, but the slight play allowed generally prevents the springs from being broken. Two sorts of set screws are used : one with a spring, as in the upper figure ; and one without a spring, as shown in the lower figure. They are generally fitted alternately round the relief ring. In the event of the spring breaking, the screw H can be pressed on the end of the spindle B, and secured in any required position by the lock-nut K. N is a gland-shaped piece, used to contain the screw H ; after the spindle and spring are fitted in place, N makes a steam-tight joint with the valve-chest cover. The set screw L is accurately measured for the required length, leaving $\frac{1}{32}$ to $\frac{1}{16}$ inch play between its end and the flange of the relief ring, and a washer M is inserted to prevent any maladjustment.

The gland at the back of the relief ring is generally made in the form of a cover for the space contained within the ring, as shown.

In Fig. 201 a somewhat different arrangement of the relief ring for a triple-ported slide valve is shown. Two wedge-shaped metal rings, with soft asbestos or other packing between them, maintain steam-tightness between the steam space and relief-ring space. A small hole connects the valve exhaust cavity with the relief space. The valve is shown in the position obtained just after the beginning of the up-stroke.

Martin and Andrews Balanced Relief Slide Valve.—In this arrangement, which is usually fitted with a double-ported type of valve, the whole of the pressure is relieved from pressing on the back of the valve by using a cage which is fitted inside the slide valve casing, as shown in Figs. 202 and 203.

Fig. 202 shows a plan of the cage which surrounds the slide valve on three sides and is pressed tightly against the false face by two or

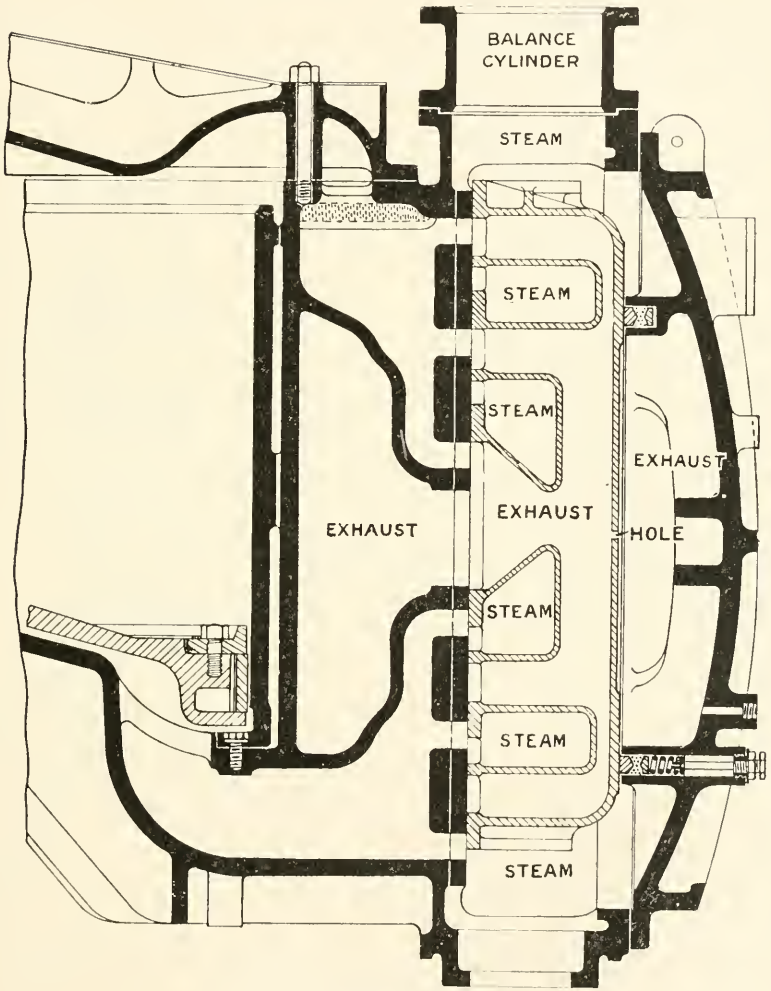


FIG. 201.—Triple-ported Slide Valve.

more coach springs S. At K there is a further relief support which only comes into play in case the springs break or do not act; in a large valve fitting a space of about $\frac{3}{16}$ths of an inch is left clear at the butts K. The cage is supported in a vertical position by abutments L

and brackets *A*. The slide valve is double faced and works inside the cage, just sufficient clearance space being allowed, front and back, and from side to side, to allow a working fit of about $\frac{5}{1000}$ths of an inch, but this varies according to size and practice.

In Fig. 203 the valve is shown at about the top of its stroke when

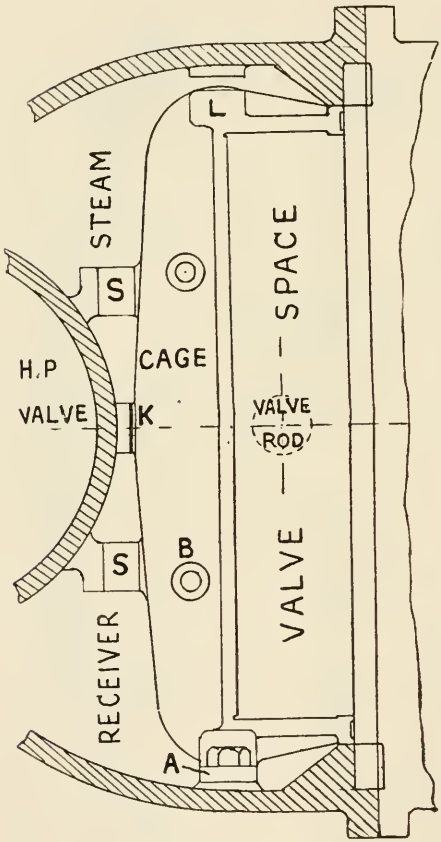


Fig. 202.—Plan.

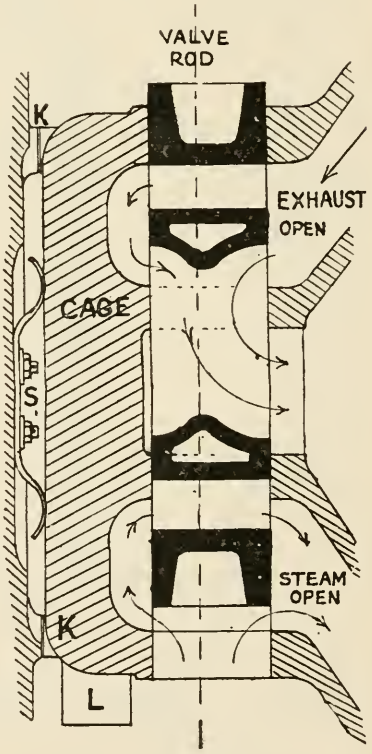


Fig. 203.—Sectional Elevation.

Martin and Andrews Balanced Relief Slide Valve.

giving maximum opening to steam below the piston, with the exhaust open to the space above the piston.

This type of valve is fitted to the M.P. engine of the *North Western Miller*; without it the slide valve would be pressed against the false face by a total steam pressure of about forty tons, and the consequent

friction between the two working faces of the valve and cylinder face would be very great. The author fitted several of these valves to merchant vessels during the war, and in every case the wear and tear was considerably decreased and frequent cost of refitting became unnecessary.

Two bosses *B* are usually provided for fitting eyebolts for lifting purposes.

Cylindrical or Piston Slide Valve.—If an ordinary single-

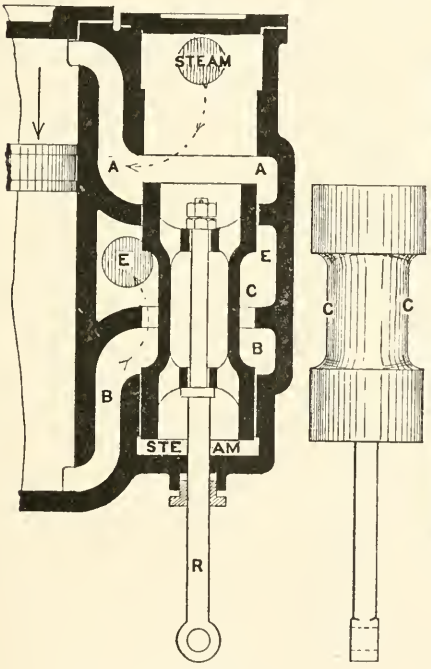


FIG. 204.
Cylindrical Slide Valve (end admission).

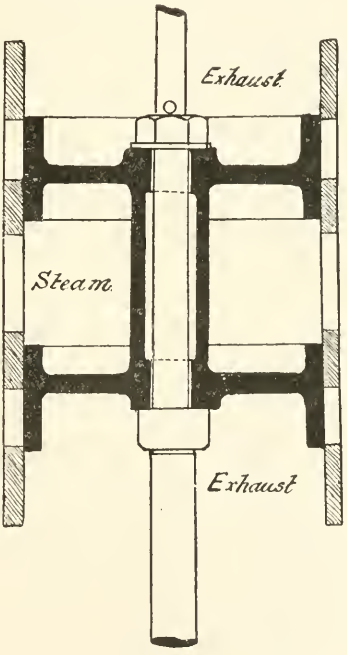


FIG. 205.
Cylindrical Slide Valve (central admission).

ported slide valve has its face curved round until it ultimately forms a cylinder, a cylindrical slide valve is produced of the type shown in Figs. 204 and 205. The length of the port is equal to the circumference of the valve, and approximately one-third the travel of the valve is required to give the same area of opening to steam compared with a flat valve whose width is equal to the diameter of the piston valve. In a flat valve the width of the port is assumed to be about equal to the diameter of the cylinder, and if the piston valve is of the same

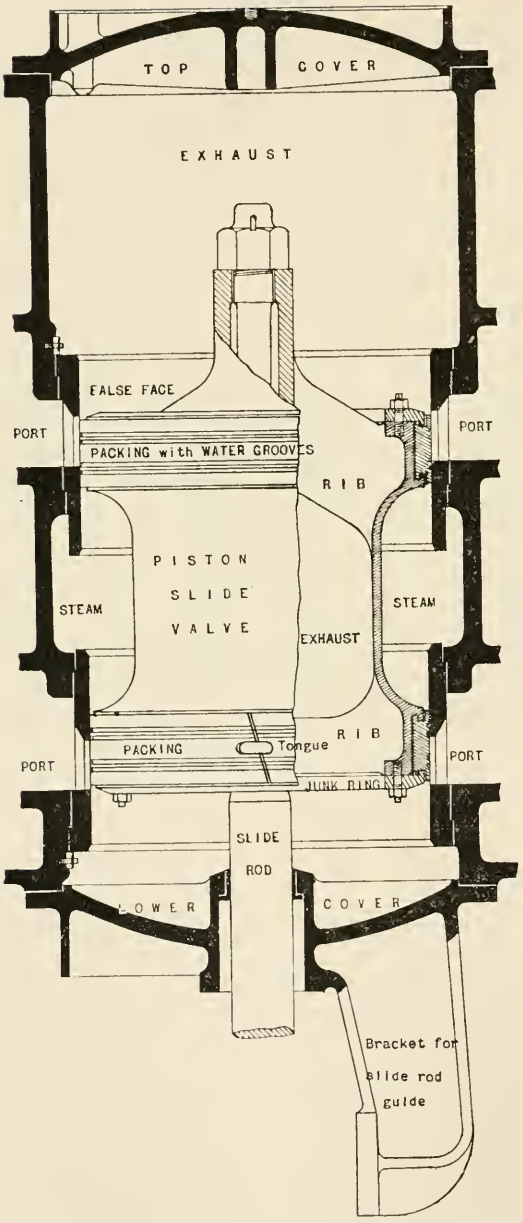


Fig. 206.—Cylindrical Slide Valve, with Packing Rings, etc.

diameter, it is evident that the fore and aft length of engine must be increased. A diameter of valve about two-thirds the diameter of cylinder is generally employed, and this also reduces the clearance spaces, which are very large. With this diameter the same travel as the double-ported valve may be used.

The arrangement of double ports is usually too complicated for practical use; but two cylindrical valves are sometimes used for the M.P. cylinder, and are then worked by the same link motion. This reduces the necessary length of engine, and has been adopted for several U.S.N. battleships.

Cylindrical slide valves are generally fitted to the H.P. and M.P. cylinders, which are comparatively small, and any leakage which may take place is not of the same importance as leakage through the L.P. slide valve, because the steam is used in another cylinder. Many small

engines are fitted with cylindrical slide valves only, and no loss from leakage is apparent.

The H.P. valve is frequently fitted without packing rings, as shown in Figs. 204 and 205, but in all other valves packing rings are generally fitted. The steam is preferably admitted round the middle part of the valve, as shown in Fig. 205, because the valve-rod packing is then only subject to the lower exhaust pressure, which tends to decrease leakage and friction.

The valve moves on a cylindrical false face, which is secured, as shown in Fig. 206, in a similar manner to that in which the liner is fitted to the cylinder. The ports are generally ribbed diagonally, to prevent the packing rings being caught in them, but it is preferable to make the rings wider than the ports, as in Fig. 206, to prevent this occurring.

The packing rings are similar to those fitted to the piston, and a junk ring is also fitted, if required, for renewing or repairing the packing rings. Fig. 206 shows a cylindrical slide valve fitted with packing, junk ring, and tongue piece. The rings are restrained ; this is usual for all packing rings fitted to slide valves.

No equilibrium or relief ring is necessary for a cylindrical slide valve, because there is no pressure on the false face except that exerted by the packing rings or springs.

If steam is admitted through the middle part of the valve, no balance piston is required to carry the weight, if the upper part of the valve can be made sufficiently greater in diameter than the lower part. When steam is admitted at the ends of the valve, a balance piston is necessary, and is generally fitted.

Balance Piston.—The weight of the slide valve and its connections, which are raised by the eccentric motion at each revolution of the engine, is very great, and the consequent friction between the eccentric straps and sheaves is objectionable. By fitting a small piston on the end of the valve rod above the valve to work in a small cylinder fitted above the valve chest (Fig. 200), the weight of the valve and its connections is supported by this piston. The upper side of the piston is connected with the exhaust from the slide valve to which it is attached, and its lower side is in connection with the steam on the back of the valve. The size of the balance piston is so proportioned that when the engine is working at full power the resultant total upward pressure on the piston is slightly in excess of the weight it has to support.

The balance piston acts as a guide for the slide-valve rod, and

thus tends to keep the valve rod in proper position and working line ; but as the hole through which the rod passes in the flat slide valve is elongated to allow the valve to be pressed against the cylinder face, this is not of much use to the slide valve except in extraordinary conditions. The lower part of the valve rod below the valve chest is generally fitted with a guide and block, so that there is only a direct lift on the link motion, as distinct from a bending action, which is otherwise produced.

Momentum Balance Piston.—This is a form of balancing arrangement, fitted in some cases where the valves are heavy, and therefore likely to bring a lot of work on the link motion when the slide valve reverses its direction of motion at each end of its stroke.

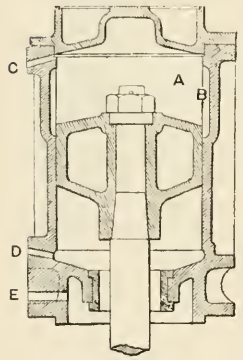


Fig. 207.
Momentum Balance Piston.

A sectional elevation of the momentum balance cylinder and piston is shown in Fig. 207. The piston is fitted on a continuation of the valve rod and immediately above the valve. The upper part of the cylinder *A* can be connected, if required, with the exhaust steam through the pipe *C*. The lower part of the cylinder below the piston is connected with the steam side of the valve through an annular space about the rod, provided by making the neck bush larger than the rod.

On the down-stroke of the valve, the steam is imprisoned below the piston by the closing of the connection at the neck ring, where a spigot or projection on the piston descends into the space about the rod. The imprisoned steam forms a cushion, and takes the momentum of the lowering weights and brings them gently to rest, and at the same time accumulates sufficient pressure to assist the reversal of direction of motion.

On the up-stroke, the steam is also imprisoned, and a cushion is formed, but the accumulated pressure is moderated by allowing some of it to escape through two grooves *B* made in the sides of the cylinder near its upper part, just previous to the top of the stroke. If the pressure above the piston is less than the steam pressure, there is access through the grooves, and sufficient enters for the necessary cushioning, which commences after the openings are closed by the upward motion of the piston.

In the event of too great cushioning and resulting shock, the space

above the piston can be connected with the exhaust by opening a cock at C; and the space below the piston can be connected by opening a cock fitted in a pipe connecting up D and E.

Joy's Assistant Cylinder.—In this arrangement (Fig. 208) the balance piston tends to relieve the link motion of all work except the initial guiding motion. For large engines and valves the assistant cylinder may develop as much as 25 I.H.P., and by the application of the indicator an exact reading of the action inside the cylinder can be made and any calculations checked. The balance piston is sometimes kept moderately steam-tight in its cylinder by

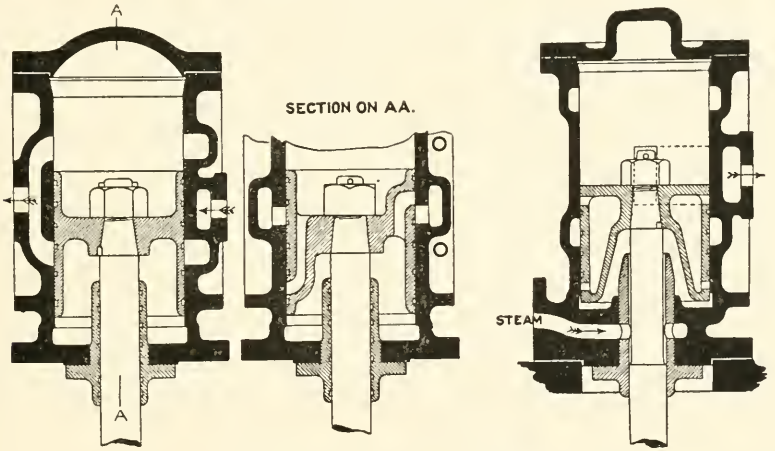


FIG. 208.—Joy's Assistant Cylinder.

two or more Ramsbottom rings, each of greater width than the steam ports which they travel across.

Steam is admitted to an annular space in the cylinder at about its mid-length; the piston overlaps this space at all parts of its stroke. Steam ports are fitted in the piston to admit steam as required either above or below it, and the opening is regulated by the movement of the piston. The connection with the exhaust is also opened by the movement of the piston, and exhaust ports are made in the cylinder, which are uncovered at the correct instant of the stroke. The steam and exhaust ports are arranged so that there is no passage-way between them when the piston is in place.

For the up-stroke, steam admission begins at about one-tenth before the end of the down-stroke, and is cut off when the piston reaches one-tenth of its up-stroke, and is expanded until the piston

uncovers the exhaust at about six-tenths of the stroke, when release begins. Above the piston the exhaust is closed at about four-tenths of the stroke, and compression then begins, and continues until admission begins for the down-stroke, at about nine-tenths of the up-stroke. For the down-stroke the actions are similar; admission (or lead), cut-off, expansion, release, and compression are all repeated.

For naval engines, which are frequently worked with the link considerably shortened, this arrangement may become inoperative

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References to Fig. 209.

A, Steam cylinder, in which piston works connected with rod *B*.

B, Piston rod, common to both pistons, steam and hydraulic.

C, Hydraulic cylinder, in which a piston works attached to the common piston rod *B*, packed with double packing leathers, which open to pressure above or below the piston, and maintain pressure tightness.

D, Crosshead on piston rod.

EE, Rods connecting crosshead with reversing arm of weigh shaft.

F, Control valve for hydraulic cylinder, worked by rod common to slide valve of steam cylinder.

G, Tank containing mixture of water and glycerine, connected with control valve and with bottom and top of hydraulic cylinder through a three-way cock.

H, Reversing lever, with spring attachment for setting lever in sector.

K, Sector.

L, Fulcrum of lever *DLM*.

M, Rod connecting the lever *DLM* with the common valve spindle *N*.

N, Valve spindle, connected with both hydraulic control valve and steam slide valve.

O, Curved link rigidly attached to lever *DLM*.

P, Block carried by crosshead *D* and sliding over link *OO*.

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if the travel of the piston is insufficient to admit steam to the assistant cylinder at each stroke.

Brown's Reversing Gear.—This arrangement for reversing the engine is fitted in a great number of ships of the mercantile marine, and the type shown is very similar to that fitted in the *Agamemnon*. It is also used for operating the manœuvring valves of the turbine engines of the *Mauretania*, etc.

The reversing gear can be moved by steam, by pulling the lever *H* into the "ahead" position, or by pushing it into the "astern" position. When required to go ahead, the fulcrum *L* is raised by

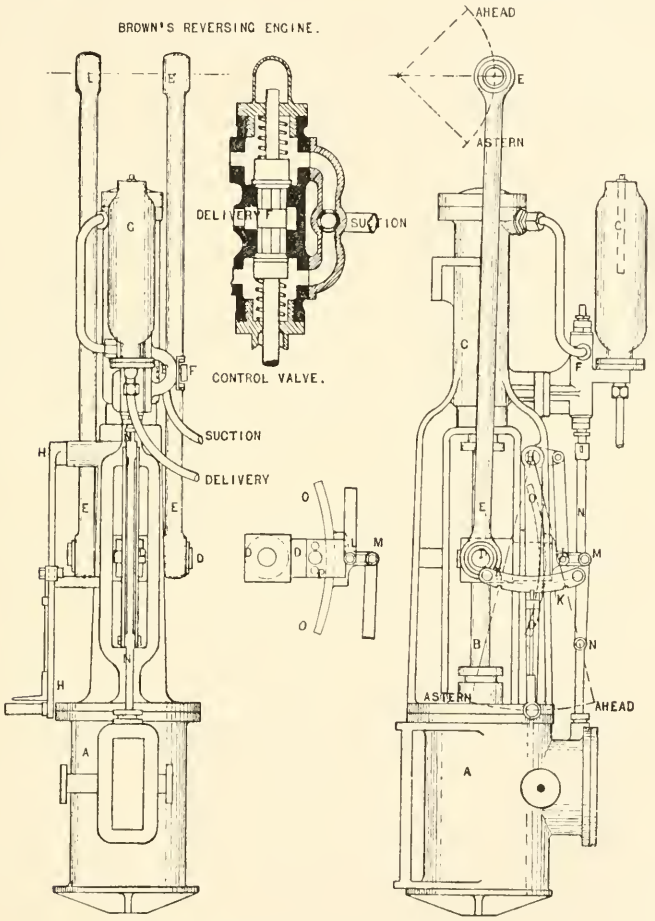


FIG. 209.—Brown's Reversing Gear.

moving the lever *H*; the connecting rod *MN* is thus moved upwards, and the slide valve with it; steam is admitted below the piston, and raises the reversing arm towards the ahead position. As the pistons rise, the block *P*, attached to the crosshead *D*, tilts the lever *DLM* about the fulcrum *L* (the vertical movement of which is restricted by

the reversing lever H), and returns the slide valve to its central position, thus bringing the steam piston to rest.

The movement of the buffer piston in its cylinder is controlled by the valve spindle N, which is also attached to the water-control valve. When the steam valve is closed, the control valve is also closed, and as no water can then pass from one side of the buffer piston to the other, any movement of the steam and water pistons is checked.

The reversing arm of the weigh shaft can be stopped in any intermediate position, for linking up, by returning the reversing lever H into the middle position as required by hand, and thus closing the cylinder port to steam and closing the control valve.

When steam is not used and hand-gear is required, the reversing arm of the weigh shaft is moved into any desired position by using a hand-pump (hydraulic) and manipulating the three-way cock. Water is drawn into the pump from the tank, which is connected through the three-way cock with the proper end of the cylinder; and by setting the reversing lever, as for steam working, pressure is admitted through the control valve to the opposite end of the cylinder, and moves the piston into the required position.

The control valve is the solid piston F on the rod between the two valves shown in the enlarged figure. The rod passes through the centre of the valves and does not move them; they are returned to their seatings by spiral springs.

Joy's Valve Gear.—In this arrangement the movement of the slide valve is derived from the connecting rod without the use of eccentrics. In Fig. 210 the connecting rod is represented by ABC, and the lever DEF operates the valve through the fulcrum E. The end F of the lever is attached by a connecting rod to the slide-valve rod, and the other end, D, is attached to the connecting rod ABC by the links BG and GH; where B is a pivot on the connecting rod ABC, and moves with it, and H is a fixed pivot on the engine framing, and is the only fixed axis in the gear. The fulcrum E of the lever DEF is constrained to move along PP', or QQ', as required for ahead or astern working. In the figure, E is suspended to a rod, the point of suspension of which rod can be altered, as shown in dotted lines. (In Hackworth's gear the curve PP' is a slot, along which the point E is constrained to move; and the inclination of the slot can be altered to QQ' for astern working. In Marshall's gear a suspension link is used, as shown in the figure, and the point of suspension is altered to obtain astern working.) Intermediate positions correspond to various degrees of linking up.

Two independent motions are given to the point F:

1. The reciprocations of the piston are transmitted to F by the action of the lever DEF. This part of the motion is constant, and independent of the position of the gear, and the proportions are chosen so that its amount is equal to the *lap* added to the *lead*. In other words, the ratio of EF to ED is made equal to the lap *plus* lead, divided by half the stroke of the piston.

2. The movements of the connecting rod ABC are transmitted to the fulcrum E, and cause it to move along the curve PP', or QQ', according to the position of the gear for ahead or astern working. This additional portion of the valve's motion, obtained from the movement of the connecting rod, provides for the port opening beyond the amount given by the *lead*, and can be regulated by altering the angle of inclination of the curve PP'. This alteration provides for astern working and for various degrees of cut-off and expansion; all the operations of the slide valve are satisfactorily obtained by this valve motion.

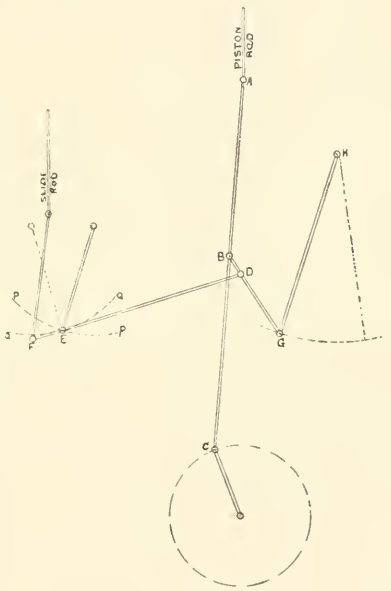


FIG. 210.—Joy's Valve Gear.

If one end of the lever FED were attached directly to the connecting rod at B, the varying obliquity of this lever itself would cause the fulcrum E to move slightly to and fro on the curve; to avoid this, and to secure that the movement of the fulcrum on the curve should be derived from the movement of the connecting rod ABC only, the parallel motion $EDBGH$ is fitted. This motion is so proportioned that if the point B be moved along BC in a straight line, the point E remains stationary; therefore the disturbing effect of obliquity of FED is avoided.

There are many other valve gears where the movement of the valve is derived from that of the connecting rod, or from one eccentric, such as in Bremme's. The objection to them generally is the number of pin joints and working surfaces, which are likely to produce lack of

uniformity in the motion of the slide valve after they are somewhat worn. The advantage apparent from fitting the slide valve in front of the cylinder, as shown in Fig. 210, is a shorter engine, with a slight saving in weight; it is more than probable that this advantage outweighs any of the objections that can be raised.

Geometrical Representation of the Motion of the Slide Valve.—Take XX' and YY' at right angles to each other, as in Fig. 211, and intersecting at A; from the centre, at radius equal to the half-travel of the slide valve, describe a circle, XCD. From any point B in the line XX', at radius BC, equal to the length of the eccentric rod, draw an arc of a circle, CED, cutting the circle XCD at C and D, and cutting AB at E. Then the point E moves in its path XX', which represents the travel of the valve, exactly as the valve, as represented by the point B, moves. (The proportion of AC to CB is much exaggerated in the figure to show the effect of obliquity more clearly.)

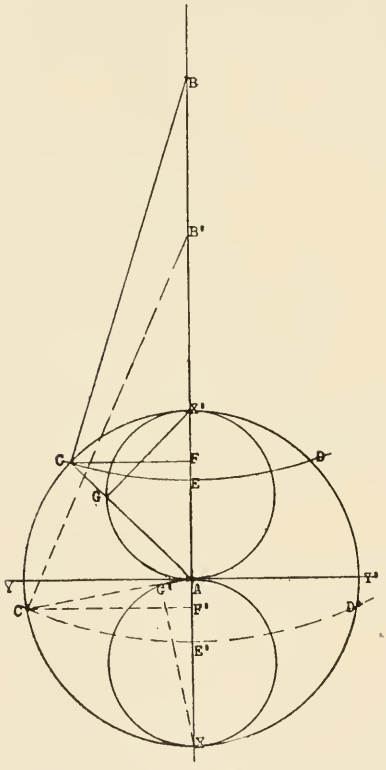


Fig. 211.—Geometrical Motion of the Valve.

The eccentric rod is generally very long compared with the eccentric radius—eight or more times; and if it be indefinitely long, the position of the valve in its path may be represented by a point, F, where CF is drawn perpendicular to AX. The length EF is very small, and may be neglected; technically, this is termed neglecting the obliquity of the eccentric rod.

Set off AG equal to AF, that is, the distance of the valve from the middle of its stroke. Join GX'. In the triangles CAF and $X'AG$, AG is equal to AF, and AC to AX', and the angle CAX' is common to both; therefore the angle AGX' is equal to the angle AFC, that is, to a right angle.

If various positions of F be laid off, it is seen that G lies on a pair of circles, having AX' and AX as diameters. Thus, the radial distance AG of any point, G, on these circles from A, represents the distance the valve is from the middle of its stroke, for the same position of the eccentric radius AC.

Next, suppose the engine crank is on one of the dead centres, AX (Fig. 212), and rotating in the direction of the arrow. Then the eccentric radius is in a position, AC, such that YAC equals the angle of advance, and the position of the valve is represented by M, the foot of the perpendicular, CM to AX. Take AG, equal to AM. Now, suppose the crank to turn into a new position, AC', through any angle, a, the eccentric radius will move through an equal angle, CAC'', and the valve moves from M to M''. Along AC' take AG'', equal to AM'', and continue the construction for any other positions.

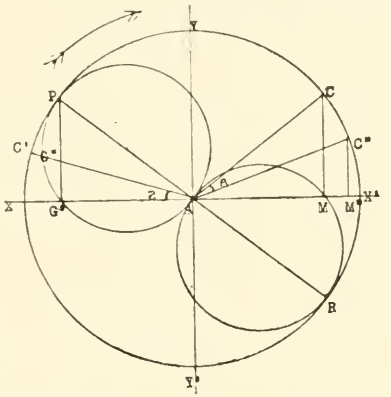


FIG. 212.—Geometrical Motion of the Valve.

Then G is found to lie on a pair of circles, each of whose diameters is equal to the half-travel of the valve, *i.e.* equal to AX. One of these two circles touches the circle XPC, whose radius is equal to the half-travel of the valve at P. Now, in the triangles $AG'P$ and AMC, AC is equal to AP, and AG' was made equal to AM, and each of the angles AMC and $AG'P$ is a right angle; therefore the angle PAX is equal to the angle CAX'. The angle PAY and angle CAY are also equal, because they are the complements of the angles PAX and CAX'. That is, the angle PAY is equal to YAC, the *angle of advance.*

Obliquity of the Eccentric Rod.—Reference to the above figures (211 and 212) shows that obliquity cannot be neglected when setting the slide rod and valve practically. In the case of a vertical inverted direct-acting engine, the tendency is to increase the mean downward pressure and to diminish the mean upward pressure; this, added to the weight, tends to produce an irregular motion of the engine.

To overcome this tendency, the slide rod is lengthened so that the centre of the valve in the middle of its stroke is a little higher than the

centre of the exhaust port in the cylinder face. This increases the steam lap at the top and diminishes it at the bottom, but the lead diminishes at the top and increases at the bottom. The sum of the two laps, the sum of the two leads, and the mean cut-off, remain as before. Thus for the general purposes of designing slide valves for various engines, and the graphical measurement of the various functions, the method, neglecting obliquity, retains all its usefulness.

Zeuner's Diagram.—All the *operations* of the valve may be represented geometrically by means of the Zeuner valve circles, shown in Fig. 213.

Take XX' and YY' at right angles to each other and intersecting at A. Through A draw PAR so that the angle YAP is equal to the angle of advance. Make $AP = AR =$ radius of eccentricity, so that PR is equal to the travel of the valve. Describe circles on AP and AR as diameters. Describe a circle with a radius to represent the path of the crank pin. Describe circles NNN, with a radius equal to the steam lap, and LLL, with a radius equal to the exhaust lap, from the centre A.

For any position of the crank AK, where AK cuts the valve circle AMP at M, the distance the valve has travelled is represented by AM. MN is the opening of the port to steam. ML is the opening of the exhaust port, but as the opening is frequently limited to that necessary for admission it may be only LE as a maximum instead of LR or LP. Similarly, AQ is the position of the crank when admission begins ;

AU	,,	,,	cut-off takes place ;
AT	,,	,,	release begins ;
and AS	,,	,,	compression begins.

If there is no exhaust lap, AT and AS will be at right angles to PAR. The corresponding elementary indicator diagram, for the same construction, is shown in the lower figure.

When linked up, a very approximate diagram of the motion and operations of the valve can be obtained by the application of the above constructions.

First construct the diagram for working in full gear (Fig. 214), where AP represents the half-travel of the valve. When the link is middled the travel of the valve is approximately represented by *twice the length of the perpendicular let fall from the centre of the shaft on the line joining the centres of the eccentrics* ; set off this length along AX. Through the point found above, and through P, draw an arc of a circle, PBP, with a suitable radius and centred on AX, as shown.

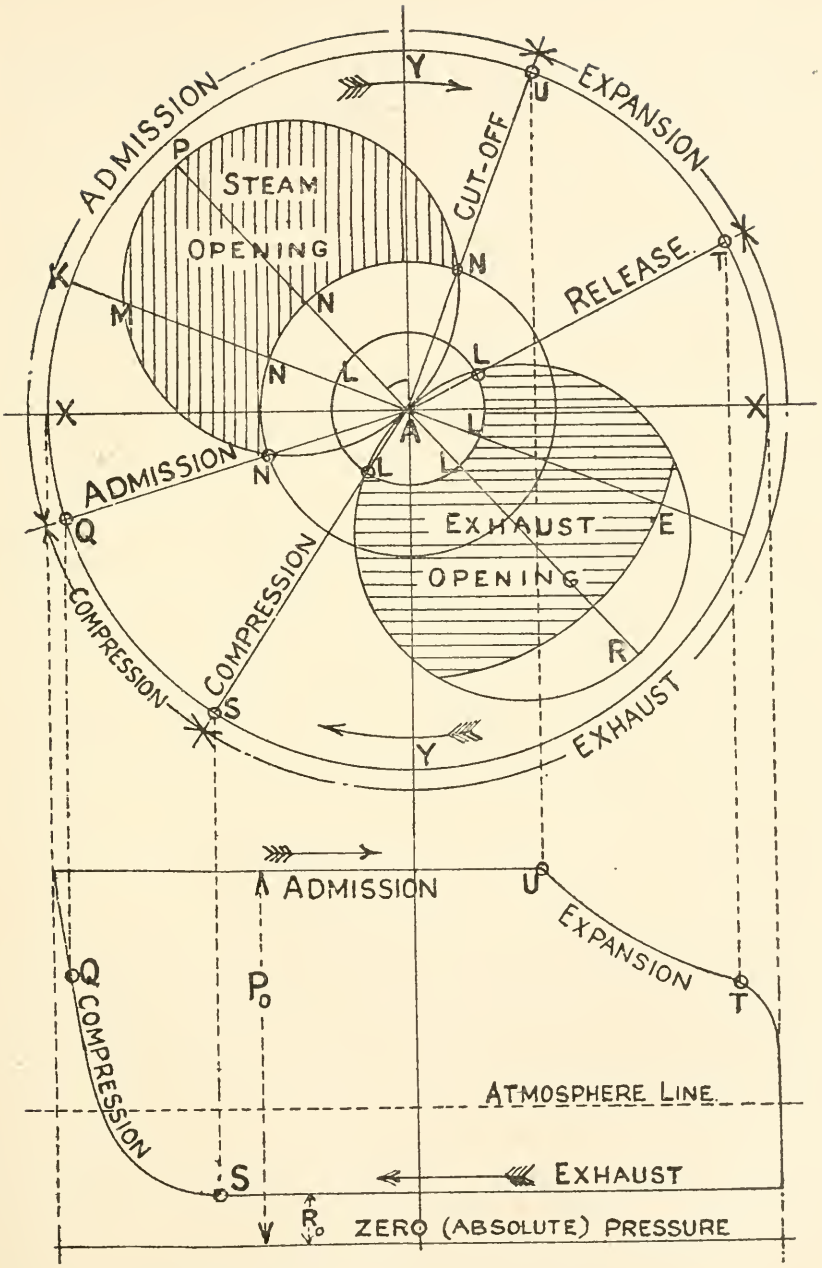


FIG. 213.—Zeuner Circles for Slide Valve Motion.

PBP represents the length of the link, and by setting off *PB* so that it represents to scale the position of the block when linked up, *AB* is the maximum travel of the valve from its central position.

In Fig. 215, suppose the block is in some position, *Z*, between *L* and *K*; then—

$$PB : PP \text{ (Fig. 214)} = LZ : LM \text{ (Fig. 215)}.$$

The construction for the valve circles is shown in Fig. 214, and also the positions for the various operations of the valve when linked up.

In cases where the link is *overhung, as with the solid-bar link,* the

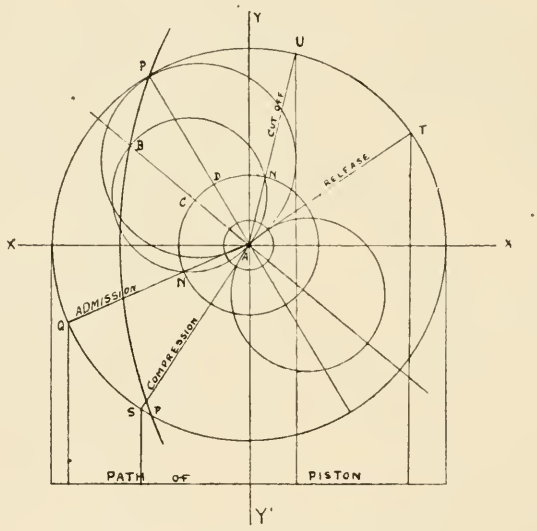


Fig. 214.—Zeuner Circles, when "linked up."

points *L* and *M* (Fig. 215) are always outside the extreme positions of the link block, and consequently the gear is always linked up. The construction of the valve circles is obtained by first constructing those theoretically due to the full travel of the eccentric radii, and then following the procedure shown above for the valve when the link is shortened.

Or, when the travel of the valve is known for both full gear and in mid-link, the curve can be found as above; call this *PP* in Fig. 214, then take a point, *D*, outside *PP*, such that—

$$\frac{PD}{PP + 2 \,.\, PD} = \frac{\text{distance between end of link and full gear}}{\text{full length of link between centres}}.$$

The diagrams shown in Fig. 215 are for " end " admission, as shown in Figs. 191, 192, 194, and 204. With " central " admission, as shown

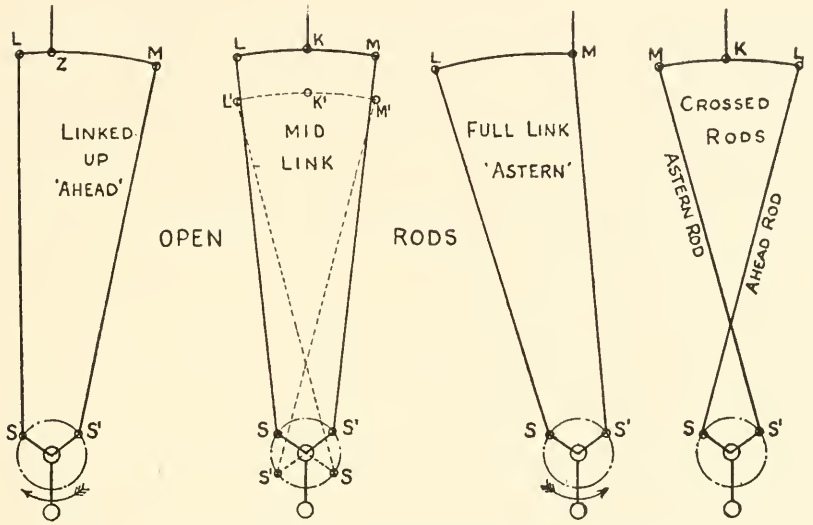


FIG. 215.—Open and Crossed Rods, with " end admission."

in Figs. 205 and 206, the crank would be on the top instead of the bottom centre with " open " and " crossed rods " respectively. Figs. 195 and 196 show an example of open eccentric rods with central admission.

PART VI

THE MARINE STEAM TURBINE

CHAPTER XX

GENERAL PRINCIPLES AND BLADING

Classification.—Turbines may be divided into two classes, according to the manner in which the steam causes their rotation.

(1) *Impulse*, *action*, or *velocity* ; in which the velocity of the steam is generated in the fixed blades or nozzles. The most notable modern examples are the Pelton wheel (water), De Laval, Curtis, Brown-Curtis, Rateau, and Zoelly.

(2) *Reaction* or *pressure* ; in which the velocity is generated in the moving blades. Hiero of Alexandria invented this type about 130 B.C., and it was used by De Laval many years ago in the form shown in Fig. 217. The best-known modern examples are the Parsons and Ljungstrom turbines, which for practical purposes are classed as *reaction* turbines, but actually some part of the velocity is generated in the fixed blading, and causes some impulse as illustrated in Fig. 218.

Combinations of these elementary classes are used in practice, of which the most common, particularly in small turbines, is the *Velocity Compounded* type, consisting of one impulse or velocity fixed ring of nozzles or blades in which the velocity and kinetic energy are generated, followed by a moving ring of blades of the impulse type, then by a fixed ring of guide blades of the impulse type, and a second ring of moving blades of the impulse type, and so on for several rings of alternate fixed and moving rings of impulse blading.

In many modern turbines of the Parsons type there are two distinct stages, beginning with a velocity compounded high pressure stage, with initial nozzles, followed by a low pressure stage of many rows of

reaction blading: this is known as the Parsons Combined Impulse and Reaction turbine. Many of the Brown-Curtis turbines fitted for a direct drive in marine turbines were of this type, but the impulse type is now generally used for the low pressure stage or stages.

There is a further classification of turbines which, being influenced by their design and construction, depends on the direction of flow of steam with reference to the turbine axis. The names are self-explanatory: parallel or *axial* flow, which includes nearly all marine turbines; *inward radial* flow; *outward radial* flow, of which the Ljungstrom is a notable example; and *mixed* (axial and radial) flow.

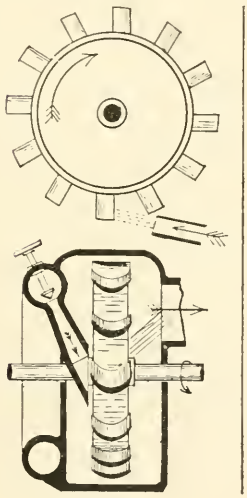


FIG. 216.
"Impulse" Blading.

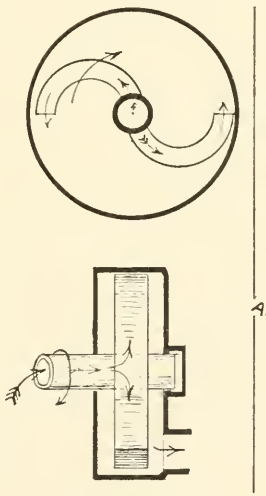


FIG. 217.
Pure Reaction Turbine.

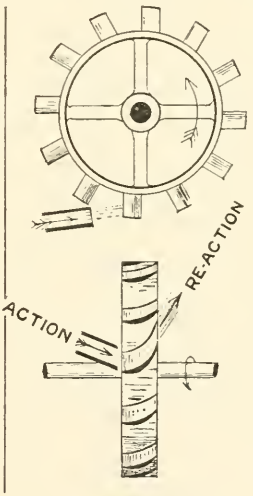


FIG. 218.
Parsons Reaction Blading.

Generally axial flow is best suited for steam and elastic fluids; and radial, or mixed, flow for water and inelastic fluids, to which the Ljungstrom turbine is the principal exception.

All turbines work on the principle of conversion of kinetic energy into work, and by this means rotating a turbine wheel or a series of wheels. The kinetic energy (energy of motion) is usually produced from some form of potential energy, such as hydraulic pressure or the heat energy of either air or steam or other gas.

As an example, consider a jet of water issuing from a horizontal pipe and striking a wall at a little distance from the mouth of the pipe or jet (Fig. 219). If the weight of water delivered per second be W,

and its velocity V in feet per second, then the force of impact or thrust or push on the wall, when stationary, evidently—

$$\text{Force of impact, or thrust} = \frac{W \cdot V}{g}.$$

Next, suppose that the wall is pushed away from the mouth of the pipe and moves with a velocity, U ; and further, that the pipe being fixed, a continuous succession of walls present themselves in front of the jet, similarly to the arrangement shown in Fig. 216. Then the change in velocity of the jet

$$= V - U.$$

$$\text{Thrust} = \frac{W}{g}(V - U).$$

Work done per second = thrust × distance moved in one second

$$= \frac{W}{g}(V - U) \times U.$$

From which the

$$\text{Horse-power (per minute)} = \frac{W \cdot U \cdot (V - U) \times 60 \text{ (seconds)}}{g \times 33{,}000}.$$

Condition for Maximum Efficiency.—It is evident, for the condition of maximum work transferred from the jet to the wall or blade, that the jet should be moving in the same direction as the blade at the instant of striking, and the fluid should not be dispersed over its

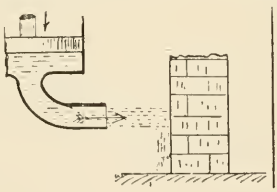


Fig. 219.
"Impulse" exerted by a Jet.

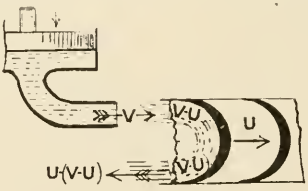


Fig. 220.
"Relative" Velocity of Jet and Blades.

surface but that its change of velocity should be effected as far as possible gradually and evenly without shock or eddymaking. This condition may be very nearly obtained by some such system as shown in Fig. 220 ; in which the blade is curved in the form of a semicircle, and the jet strikes so that the stream is exactly reversed in direction by the blade.

The *relative* velocity of the jet to the blade at the instant of striking $= V - U$; and, neglecting frictional resistances, this velocity is maintained during its motion until the stream reverses its direction and leaves the blade. Thus, at the instant of exit from the blade, the stream has a relative velocity $= -(V - U)$: because during its passage between the blades the velocity, being reversed in direction, changes its sign from positive to negative. The *absolute* velocity (relative to the earth) of the stream after exit evidently

$$= U - (V - U) = 2U - V.$$

If the whole of the kinetic energy of the jet be converted into work in rotating the wheel, the stream must have no absolute velocity on its exit from the blade ; and thus—

$$2U - V = 0, \text{ or } U = \tfrac{1}{2}V$$

and maximum energy imparted $= m \,.\, V \,.\, \tfrac{1}{2}V = \tfrac{1}{2}m \,.\, V^2$.

In other words, the condition necessary to obtain the maximum quantity of work from a jet of fluid is that the velocity of the blade must be exactly one-half that of the jet ; this is, therefore, the condition necessary to obtain maximum efficiency.

In practice, there are certain losses which prevent this maximum from being obtained ; and, stated briefly, these losses in steam turbines are caused by :—The stream not flowing into the blade passages exactly at right angles to their axis of rotation ; skin-friction of the steam coursing at high temperature through the nozzle and blade passages ; unavoidable leakages through the clearance spaces ; eddy currents due to insufficient blade velocity and errors in workmanship ; some velocity, and therefore energy, must remain on exit from the final blade passages so that the flow may be continuous ; and the exit stream does not leave the blade passages exactly at right angles to their axis of rotation.

In the " impulse " type, the steam (or other fluid) issues from a jet with great velocity and acts on a series of blades or sails standing out radially from a wheel which is revolved by the " action " of the steam. The nozzle is formed internally, as shown in Fig. 138, so that its area at first rapidly converges (corresponding to the *vena contracta*) and then diverges to areas corresponding with the adiabatic expansion of the steam combined with the velocity generated. The action inside the nozzle may be understood by imagining a bubble of steam to just fill the space at the smallest diameter of the nozzle. The bubble moves forward (like a piston) under the difference of pressure behind

and before it. As it moves forward the pressure diminishes and the steam expands to fill the rapidly increasing area, and at the same time gathering velocity due to the difference in pressure behind and before it. In this way some of **the heat energy of the steam is transformed into energy of motion of the steam, and the pressure and temperature of the steam correspondingly decrease in exact proportion to the heat converted into kinetic energy.** In designing a nozzle when expanding beyond a certain limit, actually when the ratio of expansions exceeds 58 per cent, allowance must be made for the steam to expand laterally in the nozzle, or the expansion will not be complete when the outlet is reached, and the steam will complete its expansion freely and therefore turbulently after leaving the nozzle. When the ratio of expansion exceeds the limit of 58 per cent the nozzle must be made divergent following the converging extrance already specified. Thus, in a very short, and also properly designed, nozzle it is possible for the steam on entry to the nozzle to have an absolute pressure of 150 lb. per square inch, a corresponding temperature of about 359° F., and only a very slight velocity; while at the discharge end the steam may be at an absolute pressure of 1 lb. per square inch (corresponding to 28 inches vacuum, and a temperature of 102° F.), and move with a velocity of over 4000 *feet per second.*

The total heat energy of 1 lb. weight of steam at 150 lb. absolute pressure is roughly 1190 b.t. units, and some 30 per cent of this may be transformed into kinetic energy $= \left(\frac{V^2}{2g}\right)$. It is this energy of motion which acts on the blades or sails of the turbine wheel, and a further transformation of energy takes place in creating rotation of the wheel.

Kinetic Energy of Steam.—In considering the conversion of energy which takes place in a properly constructed convergent-divergent nozzle, (Fig. 138), it is fairly evident that the expansion is approximately adiabatic; that is, little or no heat is added to or taken from the steam during its passage through the nozzle. This is particularly true of a short nozzle, such as fitted in De Laval and other turbines; but, due to radiation and other losses, is not quite true of a Parsons Reaction turbine in which the nozzle (Fig. 221), made up of successive rings of blade-passages, is very long.

By neglecting any loss which may be occasioned by radiation, etc., and considering the expansion to be adiabatic, the law of the conservation of energy states that the total energy in each 1 lb. weight of fluid at one end of the nozzle must be exactly equal to that at the other.

If H = total heat in 1 lb. of steam at the high pressure end,
and h = ,, ,, ,, low pressure end,

then, evidently, $$H = h + \frac{W \cdot V^2}{2g};$$

or, kinetic energy $= H - h = \frac{V^2}{2g}$ for 1 lb. weight.

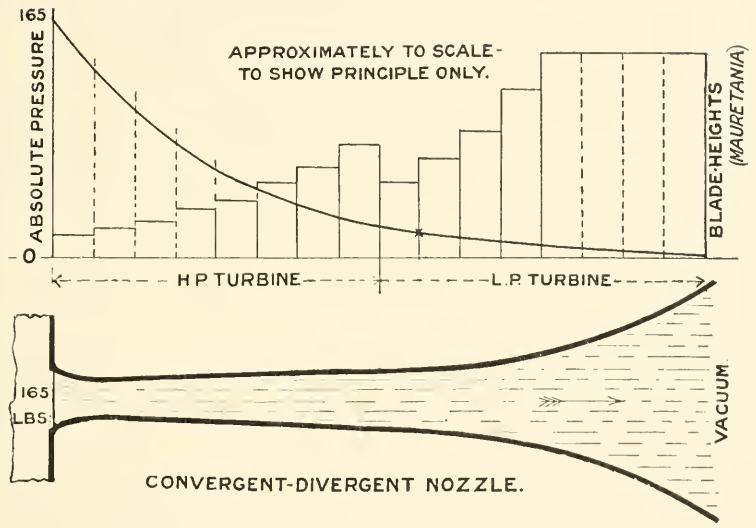


FIG. 221.—Convergent-divergent Nozzle (Reaction Turbine).

If the steam at the inlet end of the nozzle is dry saturated steam, then, according to Professor A. Rateau—

Kinetic energy per 1 lb. weight (W = 1), in b.t. units.

$$H - h = (T_0 - t_0)\left\{\frac{L}{T_0} + \frac{C(T_0 - t_0)}{T_0 + t_0}\right\}$$

Or, if the velocity is required, then

$$\frac{V^2}{2g} = 780\,(H - h) \text{ in ft. lb.}$$

where

C = specific heat of liquid (water = 1);
T_0, t_0 = absolute temperatures;
L = latent heat of saturated steam at temperature T.

The Rateau formula is correct within 1 or 2 per cent for initially saturated steam and purely adiabatic expansion.

By using the Rateau formula, the velocity can be found corresponding to adiabatic expansion between any two pressures or temperatures, and the value of $H - h$ or the available work (in b.t. units) can be determined. H can be found in any steam tables (see end of Chapter II.) and thus the value of h can be calculated.

Another method, and the most accurate for the calculation of $H - h$, is to use as a basis the formula for adiabatic expansion—

$$P \cdot V^{1 \cdot 035 + \frac{x}{10}} = \text{Constant};$$

in which x is the dryness fraction of the steam at the nozzle entry. From this formula, the percentage (z) of moisture in the steam at the delivery end of the nozzle can be calculated and the value of h determined from the usual formula—

$$h = S + (1 - z)\ L.$$

A few values obtained in this way are :

z equals 26	per cent	when expanding dry saturated steam from	250 to 1 lb. abs.
z equals 22·3	per cent	,, ,,	100 to 1 lb. abs.
z equals 16	per cent	,, ,,	20 to 1 lb. abs.
z equals 4	per cent	,, ,,	215 to 125 lb. abs.

No formula of simple character appears to cover such conditions and values, and therefore diagrams and curves have been constructed for easy reference of which Mollier's and Foster's are perhaps the best. Such curves unless of very large scale and great accuracy in delineation are not very useful, and for ordinary practical purposes the annexed table is shown.

[TABLE

Table

Available Work in b.t.u. per 1 lb. of Saturated Steam; and corresponding Nozzle Steam Velocity, in feet per second.

Initial Abs. Pressure, lb. per 1 sq. in.	Available Work (b.t. units), with Back Pressure, abs. per 1 sq. in.				Corresponding Steam Velocities, with Back Pressure, abs. per 1 sq. in.			
	14·7 lb.	2 lb.	1·5 lb.	1 lb.	14·7 lb.	2 lb.	1·5 lb.	1 lb.
5	..	70	80	104	..	1872	2000	2282
6	..	78	90	114	..	1970	2120	2394
7	..	86	99	123	..	2073	2215	2462
8	..	93	108	132	..	2156	2327	2563
9	..	100	115	139	..	2238	2395	2641
10	..	106	122	146	..	2305	2472	2704
11	..	111	128	152	..	2349	2539	2742
12	..	116	134	158	..	2398	2575	2819
13	..	121	139	163	..	2462	2641	2841
14	..	126	144	168	..	2514	2686	2900
15	..	130	149	173	..	2553	2738	2934
16	..	134	153	179	..	2592	2748	2966
17	..	138	157	181	..	2630	2797	2997
18	..	142	160	183	..	2665	2838	3021
19	..	145	163	186	..	2696	2841	3042
20	20	148	166	189	1000	2729	2886	3065
25	37	161	179	200	1365	2840	2997	3165
50	87	206	221	241	2050	3212	3333	3469
75	114	232	247	268	2394	3402	3514	3658
100	136	250	266	287	2615	3535	3647	3804
125	152	265	287	301	2740	3640	3757	3872
150	166	277	293	312	2886	3713	3817	3932
175	179	288	303	322	2995	3804	3883	4000
200	189	297	312	330	3065	3849	3932	4060
225	198	305	320	337	3155	3915	3994	4128
250	203	311	326	342	3189	3929	4030	4138

TURBINE PRINCIPLES AND BLADING

The above table of steam delivery velocities indicates the advantage obtained from a high vacuum. A difference of about $5\frac{1}{4}$ lb. from 20 lb. per square inch to atmosphere produces a velocity of 1000 feet per second, while a difference of only $3\frac{1}{2}$ lb. from 5 to $1\frac{1}{2}$ lb. absolute pressure produces 2000 feet per second, or double the previous amount. By increasing the initial absolute pressure to 250 lb. per square inch and discharging into a back pressure of $1\frac{1}{2}$ lbs. abs. a velocity of 4000 feet per second is obtained. It should be noticed that the velocity varies as the square root of the available work, which in the three above

instances are 20, 80, and 326 respectively, or approximately an increase of four times in each instance ; the square root of four is two, and the velocity is doubled therefore.

In the above table the steam is assumed to be *dry saturated*, but in practice, in the absence of superheating, by the time it reaches the nozzles it contains some moisture from radiation and condensation. The available work, in b.t.u., is therefore less in each 1 lb. weight and a deduction must be made. Assuming 1 lb. back pressure, the usual amount, the heat to be deducted for each 5 per cent of moisture varies from 5 b.t.u. at 5 lb. abs. pressure ; 10 b.t.u. at 30 lb. pressure ; 12½ b.t.u. at 100 lb. ; 13½ b.t.u. at 150 lb. ; 14 b.t.u. at 200 lb. ; to about 14½ b.t.u. at 270 lb. abs. pressure per square inch.

The presence of water in the steam supplied to the turbine, whether caused by priming or condensation, increases the frictional resistance of the nozzles and blading, retards the velocity of passage, and reduces the mechanical efficiency. Priming in small quantity is not as dangerous as in the reciprocating engine ; although the turbine is reduced in speed and efficiency. Heavy priming sometimes reduces the power and speed very considerably, and unless immediate precautions are taken in easing up the engine steam valve, damage to the turbine blading may occur. Many instances of heavy priming came under the author's notice during the war, but although the speed had had to be reduced to about *one-fourth* the full speed, only one case of serious damage caused by priming necessitated blading repairs. In this instance the H.P. blading was almost entirely stripped and melted. Lumps of metal had fused together into small globes of the size of peas and walnuts. The L.P. blading was not seriously damaged.

Superheating.—The superheat given to the steam first dries the steam before superheating it and there is no rise of pressure ; in the nozzles it counteracts condensation and other losses enumerated above ; in the turbine it counteracts condensation caused by conversion of heat energy into kinetic energy and mechanical work. Any superheat added to the steam in excess of the losses above mentioned is not likely to improve the efficiency, and the economical limit of superheat temperature as a rule does not exceed 120 to 140° F. above that of the steam in the boiler.

In some turbines with long reaction blades particularly, superheating necessitates an increased radial clearance at the blade tips to provide for the uncertain expansion in blade length under varying conditions of working, but the better regulation of superheat and better

design of blading overcome this objection to a large extent, but it should receive special consideration in design.

When the steam is superheated the quantity of superheat should be added to the available work in b.t.u. (given in the Table for saturated steam), and the steam delivery velocity can then be calculated from the formula :

$$\text{Velocity} = 224 \cdot \sqrt{\text{Total available work in each 1 lb. of steam.}}$$

EXAMPLE 1.—Assume 250 lb. abs. initial pressure, the now common practice, and 1 lb. back pressure. If the steam is *superheated* 100° F. the available work is increased by 48 b.t.u. above that of dry saturated steam, and, referring to the table, the available work in each 1 lb. of steam is—

251 358 374 390 b.t.u. with corresponding velocities of

3548 4236 4330 4420 feet per second.

EXAMPLE 2.—With the above pressures and *dry saturated* steam the figures are—

203 311 326 342 b.t.u. with corresponding velocities of

3189 3929 4030 4138 feet per second.

EXAMPLE 3.—With 5 *per cent of moisture*, not considered excessive without superheating, the figures are—

194 298 312 328 b.t.u. with corresponding velocities of

3100 3850 3930 4030 feet per second.

These examples show that there is a maximum possible gain by superheating of 100° F., when the back pressure is between 1 lb. and 2 lb. absolute, of about 14 per cent over dry saturated steam, up to 20 per cent over common steam as usually supplied when superheaters are not fitted and used. Of this 14 to 20 per cent about 10 per cent is realised in practice. If it be taken into consideration that only about 30 per cent of the heat energy of saturated steam can be transformed into kinetic energy, the superheat adds a very large part to this transformation.

De Laval Impulse Blading.—The turbine is of the "impulse" type, and the energy developed in the nozzles is imparted to a rotating wheel and ring of blades. The number of nozzles may be one or more, and in recent types the nozzles are not arranged at equal intervals around the wheel circumference but are concentrated into groups of nozzles for the larger-powered turbines. Fig. 222 shows a section through a De Laval nozzle, *B*. Steam is regulated by a valve, *A*, and

enters an annular casing, C, from the boiler. The blades of the rotor wheel are concave towards the jet, and the horns of the cavity are in axial line with the wheel. The jet is necessarily fitted, when at the side of the wheel, at some angle with the axis other than a right angle, and is fixed usually at 20° (as shown) to the face of the wheel. A section

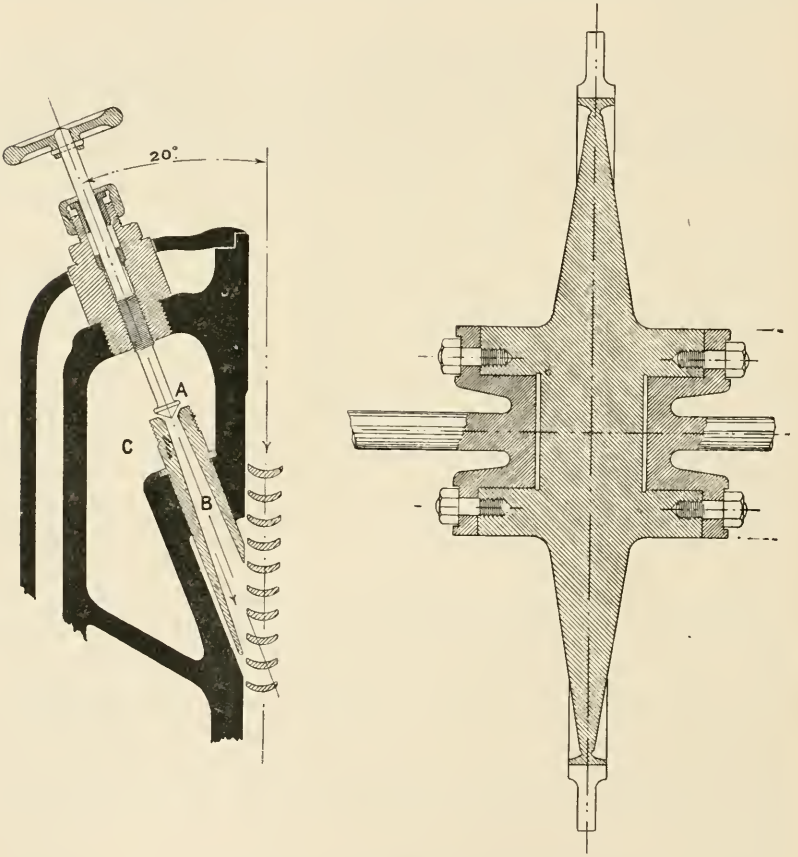


FIG. 222.—De Laval Nozzle, and Rotor Wheel.

of the rotor wheel is shown in the right-hand figure, and is of very thick construction near the axis, where the centrifugal stress is very great owing to the great velocity of the ring of blades.

In all turbines it is necessary to secure the blades very strongly to the rotor because the centrifugal stress is very great. Thus in a 300-horse-power De Laval turbine the centrifugal force of a single blade,

weighing only 250 grains or 0·0357 lb., is 1680 lb. when the wheel is running at its standard speed of 7500 revolutions per minute.

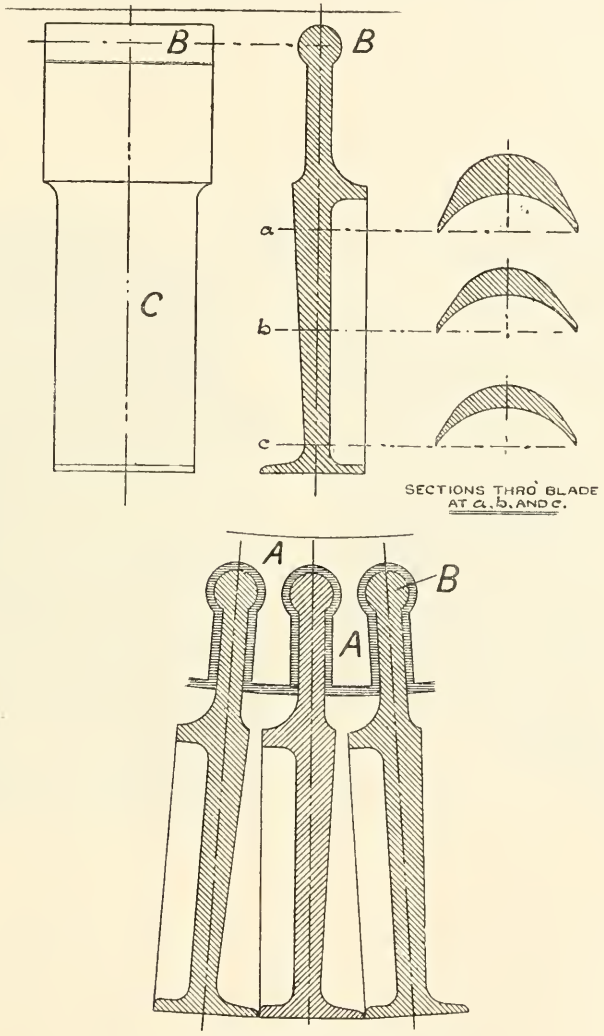


Fig. 223.—De Laval Rotor Blading.

Views and sections of the blading are shown in Fig. 223. In the rim *A* of the wheel disc, a number of holes are drilled of the correct size and at accurate distance from each other. A part of the rim is then

slotted away radially between each hole and the outer edge of the wheel. The blades are made separately and, being accurately machined, the root B is thrust into one of the recesses and then lightly caulked ; by this means each blade is fixed firmly, but can be easily removed. The

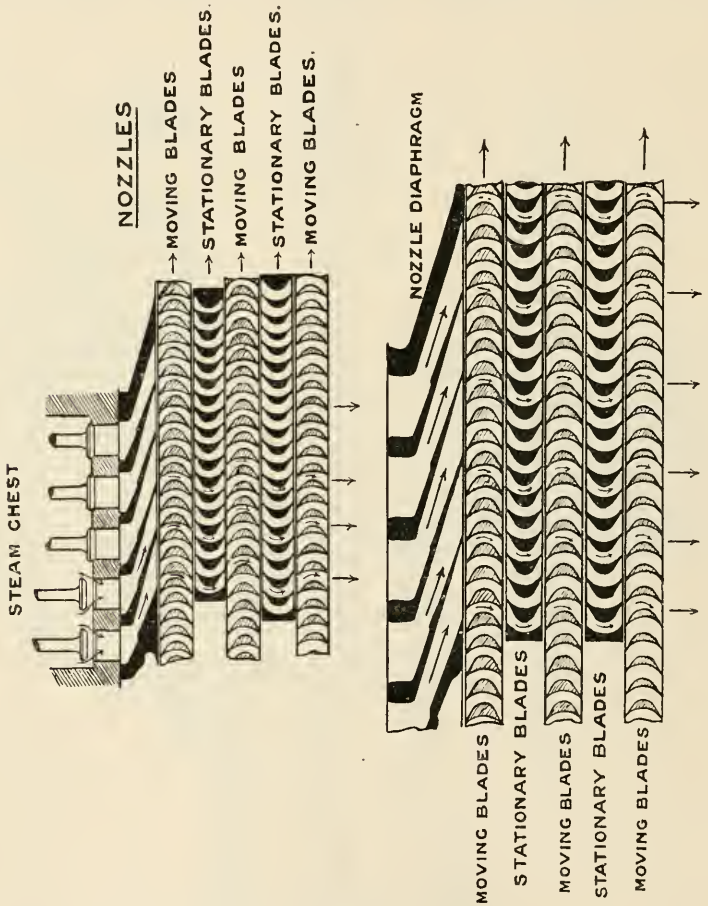


Fig. 224.—Diagram of Curtis Nozzles, Valves, and Velocity Compounded Blading.

bulb-shaped root B prevents the detachment of the blade by centrifugal force. The relative width of the blade is shown at C, and its thickness and shape at various sections by the sections, a, b, and c, and it should be noticed that each blade is much thicker at the root than at the tip. For the same reason, counteracting the effect of

centrifugal force, the rotor wheel is made much thicker near the shaft, and in the section of the wheel shown in Fig. 222 it has been considered inadvisable to drill the wheel for the shaft to pass through, and the construction shown is the result.

Curtis Impulse Blading.—The Curtis turbine is of the " impulse " type. Steam is admitted to the first ring of moving blades, shown diagrammatically in Fig. 224, through valves and nozzles of the De Laval type. After acting on the first ring of moving blades, the steam

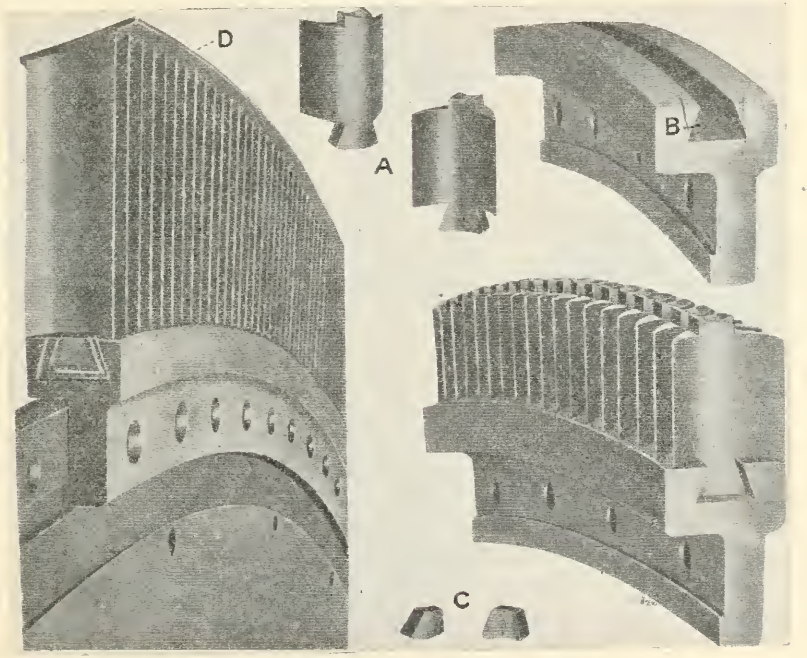


FIG. 225.—Curtis Blade Attachment.

is redirected at constant pressure by a ring of stationary blades on to a second ring of moving blades, and for large turbines is again redirected, and so on for three or four rings of moving blades until the first stage of expansion is completed and the steam escapes into a space containing a diaphragm pierced with several nozzles arranged in groups. This nozzle diaphragm constitutes the beginning of a fresh stage of operation. Kinetic energy is again generated by a further fall of pressure in the nozzles, and energy of motion is imparted to successive rings of moving blades and the steam redirected by interposed rings of stationary blades.

The steam escapes into a second chamber, ready for a third stage of expansion during its passage through nozzles fixed in a diaphragm as before. And so on for any number of stages as required, but in the final stages reaction blading was sometimes fitted on the Parsons system of continuous expansion. This Curtis system thus consists of several pressure stages with one or more rings of moving blades in each

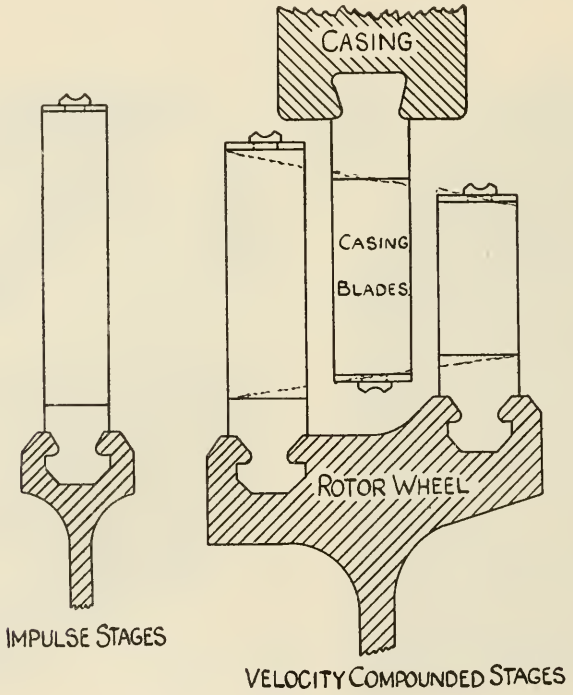


Fig. 226.—Brown-Curtis Blading.

stage. The number of stages and moving rings is based on the velocity of blade and rate of revolution.

Fig. 225 shows the method of fitting the blades, each of which in modern practice is machined from the solid *including the filling piece*, to the correct shape *A*, and they are then assembled in a dovetailed groove *B*. The outer ends are held by a shroud ring *D* and riveted over at the extreme end of the tip. Parts of rings, making up a complete circle, are riveted to a central disc, as shown in the figure. Stripping of blades is almost unknown with this method, and non-contact between

the fixed and moving parts is ensured by the large tip clearance which can be allowed without reduction of efficiency.

The Brown-Curtis blade attachment is similar to that described above. Two methods are shown in Fig. 226, one for a single ring impulse stage and the other of a two-ring velocity compounded stage. Generally, the material is supplied in long lengths of about 10 feet, of a section suitable for the blade and filling piece combined. The required length is cut off and the shape of blade and root is milled in

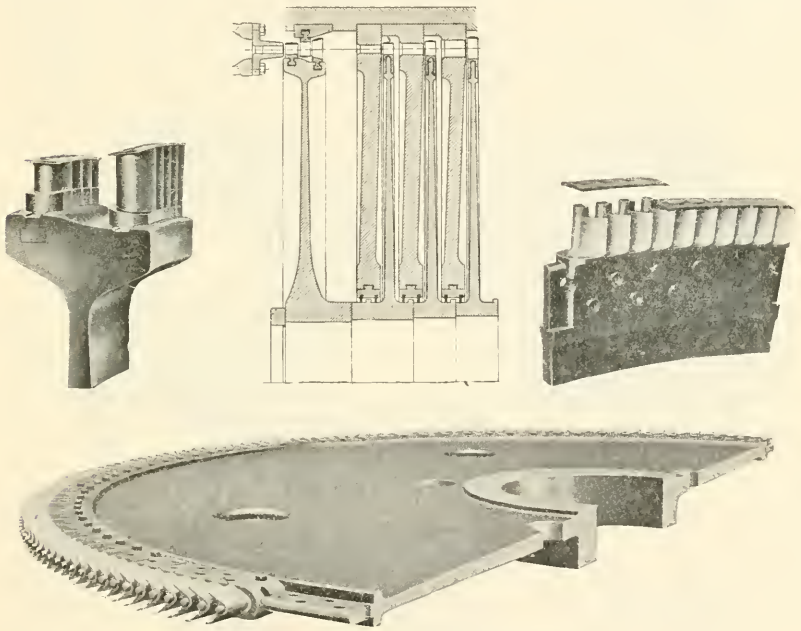


FIG. 227.—Rateau Impulse Blading.

one solid piece. The rotor or casing, as required, is grooved as shown in the sketches, and the blades are inserted through a slot, cut in connection with the groove, and shaped to suit the width of the blade root; after all the blades have been inserted and closed tightly together, this slot is closed by a well-secured stop. It should be noticed that the casing blading groove differs in shape from that in the rotors, which is arranged for resisting centrifugal action. In Fig. 226, the passage between the shrouding ring and filling piece is parallel to the axis for each blade; but in many impulse turbines, these are inclined to obtain a divergent nozzle effect, as shown by the dotted lines.

Westinghouse-Rateau Impulse Turbine. — Each blade is machined out of the solid bar and for each wheel the blades are of

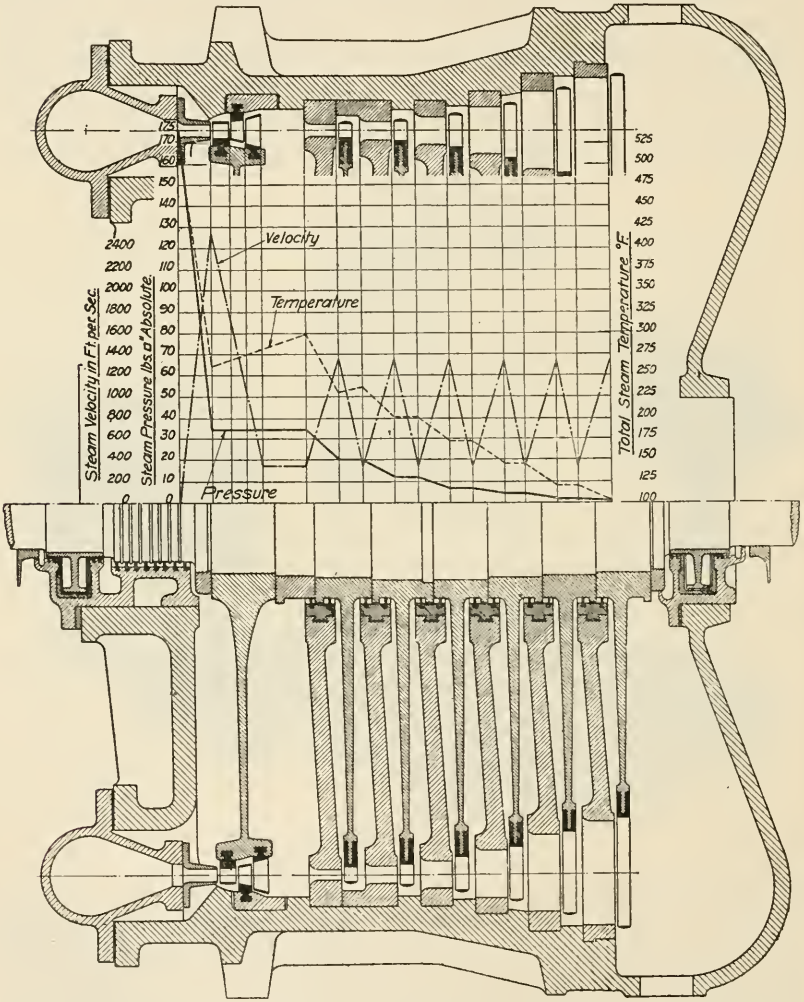


Fig. 228.—Steam Pressure, Temperature and Velocity, Westinghouse-Rateau H.P. Turbine.

exactly the same shape and weight. Two methods of fixing the blades are used, as shown in Fig. 227. The first wheel with two or more rows of blading has the same number of grooves in which the blades are passed in through two ports which are eventually closed. Where

only one row of blades is fitted to a wheel, as in the later stages, the Rateau " straddle " formation is used as shown in the right-hand view. The blades are secured by rivets in double shear and are made of 5 per cent nickel steel.

Fig. 228 shows a section through a turbine of this type designed for an initial pressure of 175 lb. abs., superheated about 155° to a total temperature of 525° F., and expanding to 1 lb. abs. In the first set of nozzles the pressure is reduced to 35 lb. abs. and the temperature to 265°, leaving a superheat of about 5° before admission to the cylinder proper and the second row of nozzles or diaphragm. The fall of pressure and temperature and consequent

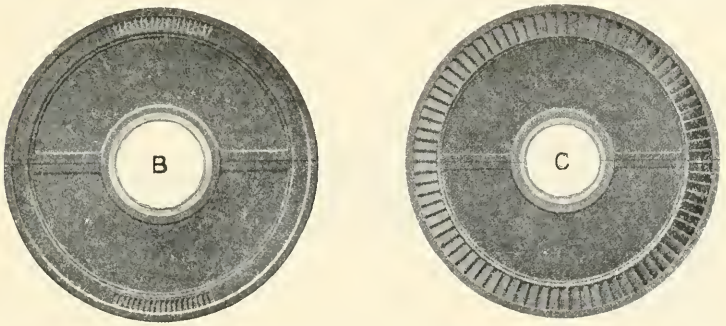


FIG. 229.—Westinghouse-Rateau Diaphragms.

fluctuation of velocity can be easily followed from the diagrams and makes an interesting study.

The nozzles are carefully designed to suit the large pressure drop, and they are supported only at the inlet end so as to allow the outlet end freedom for expansion and contraction.

Diaphragms, instead of nozzles, as shown in Fig. 229 are used for all stages except the initial nozzle stage. These diaphragms are made of cast iron or steel, with special guide blades or nozzles cast in ; they are made in halves to suit the casings into which they fit and are provided with a spigot and socket to prevent leakage from one side to the other. Partial admission, as shown in the left-hand figure, *B*, is used in the earlier stages in which the relative volume of the steam is small, and the final diaphragm, *C*, allows full admission through a complete ring of guide blades. With partial admission a greater height of blading can be used, and this gives a higher efficiency.

The *Ahead* section nozzles are usually arranged in three groups,

each group controlled by a separate valve. Each group has a different number of nozzles so that by separate or combined admission throttling losses can be avoided at seven different speeds or powers.

The *Astern* section consists of one group of nozzles only, as shown in Fig. 230, and about 70 per cent of the full ahead power can be obtained for astern steaming which is considered sufficient for practical purposes.

The *axial* clearance between the blades and stationary parts varies from $\frac{1}{8}$-inch in the smaller to $\frac{3}{16}$-inch in the larger sizes, and the *radial* clearance between the tips of the revolving blades and the cylinders is never less than $\frac{1}{4}$-inch.

Parsons Reaction Blading.—About 1894 the sectional form of Parsons reaction blading was altered from a straight passage to the

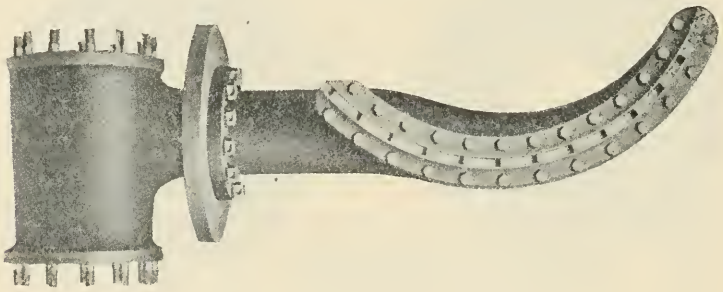


Fig. 230.—Westinghouse-Rateau Astern Nozzles and Box.

curved form shown in Fig. 231, which shows the tips of about full size for small- and medium-powered turbines. This alteration increased the Ahead efficiency, which was previously extremely low, but at the same time practically excluded reversal, even with a low efficiency.

In the general construction there are a considerable number of rings of blades fitted on a hollow cylinder called the *rotor* which forms the rotating part or working part of the turbine. Alternate with these moving rings, which stand out radially from the rotor cylinder or drum, there are a similar number of rings of blades fitted in the outer casing or fixed cylinder ; these blades stand radially inward so that their tips nearly touch the moving drum between the moving blades.

The Parsons turbine is of the " reaction " type, and therefore the blades do not stand exactly in an axial line, but somewhat inclined to it, as shown. This inclination varies in different turbines, and in nearly all those recently constructed the inclination is gradually decreased for

the last few rings near the low pressure or exhaust end. Referring to the diagram, it is seen that the first ring is fixed (fitted in the outer cylinder), and guides the steam on to the first ring of moving blades (fitted on the rotor).

A triangle of velocities shows the general action ; the steam enters with absolute velocity, and has relative velocity in the moving blade passages. The *backs* of the moving blades are shaped, on entry, so as to lie parallel to this relative velocity. The direction of motion of the steam changes within the passage between the moving blades.

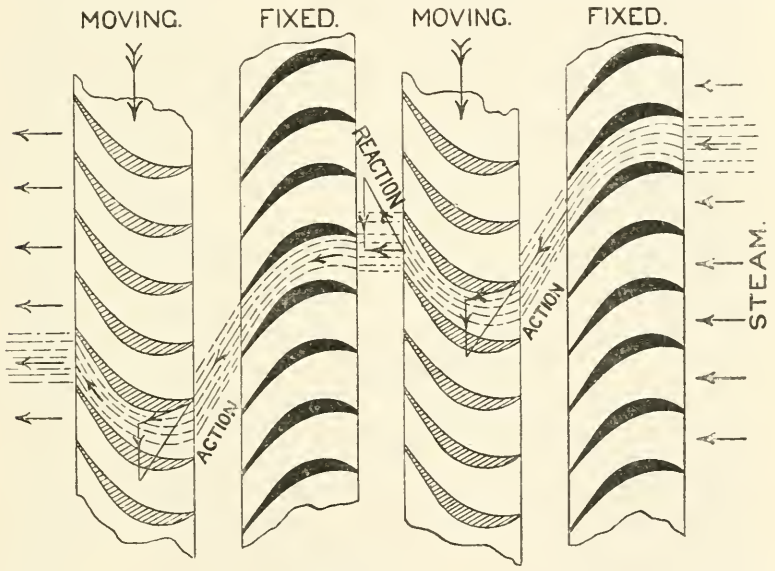


FIG. 231.—Course of Steam in Parsons Reaction Turbine.

and the steam leaves the moving passage with relative velocity, which reacts on the blades. Immediately on leaving the moving passage, the steam moves in almost an axial direction into the second ring of fixed or guide blades, which are set so that the steam enters them without shock ; the face of each guide blade is shaped so that on entry the steam travels almost in an axial direction, and the change in direction necessary for the second ring of moving blades is effected almost entirely within the guide blade passages.

There are a large number of rings of blades in a Parsons " reaction " turbine, and consequently there is only a small drop (seldom greater

that 2 lb. per square inch) from fixed ring to fixed ring. The total difference in pressure between the inlet and exhaust produces the kinetic energy which constitutes the impelling force, and, the fall in pressure being divided into stages, only a proportionate quantity of the total energy is developed and utilised in each. Thus, compared with the development of the total energy in a single stage (as in the De Laval single-wheel turbine), the velocity of the periphery of the rotor and moving blades is moderate. This produces a moderate rate of revolution, but the application of a multi-expansion turbine to marine propulsion, without speed-reducing gear between the rotor and the propeller, necessitates a very great number of rings of blades.

Referring to the *ahead* turbine *K* in Fig. 239, it will be noticed that the blades are longer as the steam approaches the exhaust end. (Each of the sets of blades of equal length is called an *expansion*.) These expansions allow for the greater volume of the steam obtained by the fall in pressure as the steam expands and does work. A diagram of the relative blade heights for the *Mauretania* is shown in conjunction with Fig. 221. The H.P. turbine rotor is 96 inches in diameter, and the blades increase in length from $2\frac{1}{2}$ to 12 inches. The L.P. turbine rotor is 140 inches in diameter, and therefore the annular passage area being greater, the blades at the high-pressure end, where the exhaust from the H.P. turbine enters, only need to be 8 inches long. At the exhaust end the blade length is 22 inches, with a tip clearance of $\frac{8}{100}$-inch. The astern rotor drum is 104 inches in diameter.

Fig. 232 shows some of the rotor blading of a Brush-Parsons turbine, in which the tips of the blades are laced together and secured to a hoop, or shrouding, ring partly embedded in the entry edges of the blades near their tips. Long blades are fitted with two or more hoop rings and lacings; this system was very common for long blading near the exhaust end of turbines. The tips of the blades are gouge pointed, so that when working if they come in contact with the walls of the rotor drum or casing the points offer small resistance and grind away without fracture.

Fig. 233 shows some of the rotor blading fitted with channel shrouding instead of the hoop rings. This shrouding is very commonly used, and instead of gouge pointing the edges of the shrouding are fined to a knife edge.

In more recent practice the shrouding shown in Fig. 234 is used, but the shrouding projects considerably over the blade tips (see left-hand view in Fig. 227), and the edges are fined so as to grind away when

they come in working contact. This system allows a very fine axial clearance and prevents clearance losses.

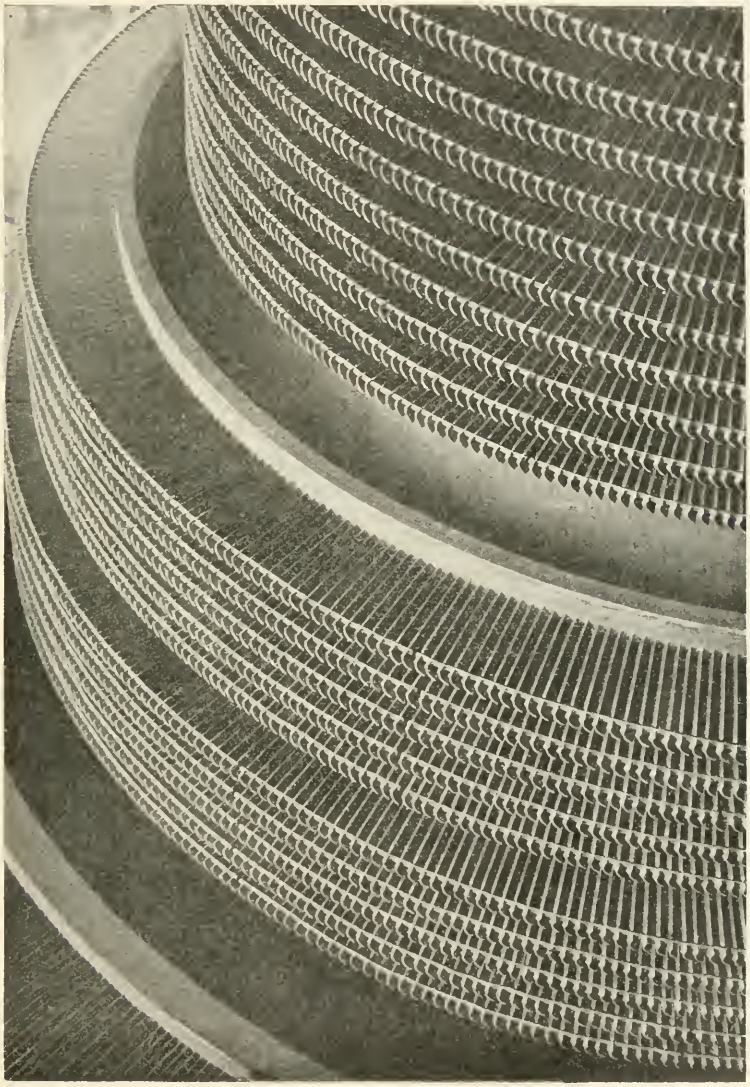


FIG. 232.—Rotor-blading of Brush-Parsons Turbine.

Some idea of the length of blading, or blade heights, may be gathered from the details of the *Mauretania*. At 194 revs. per minute, about

$26\frac{1}{2}$ knots, the surface of the H.P. moves at about 80 feet per second and that of the L.P. rotor at about 116 feet. Boiler pressure 180 lb.

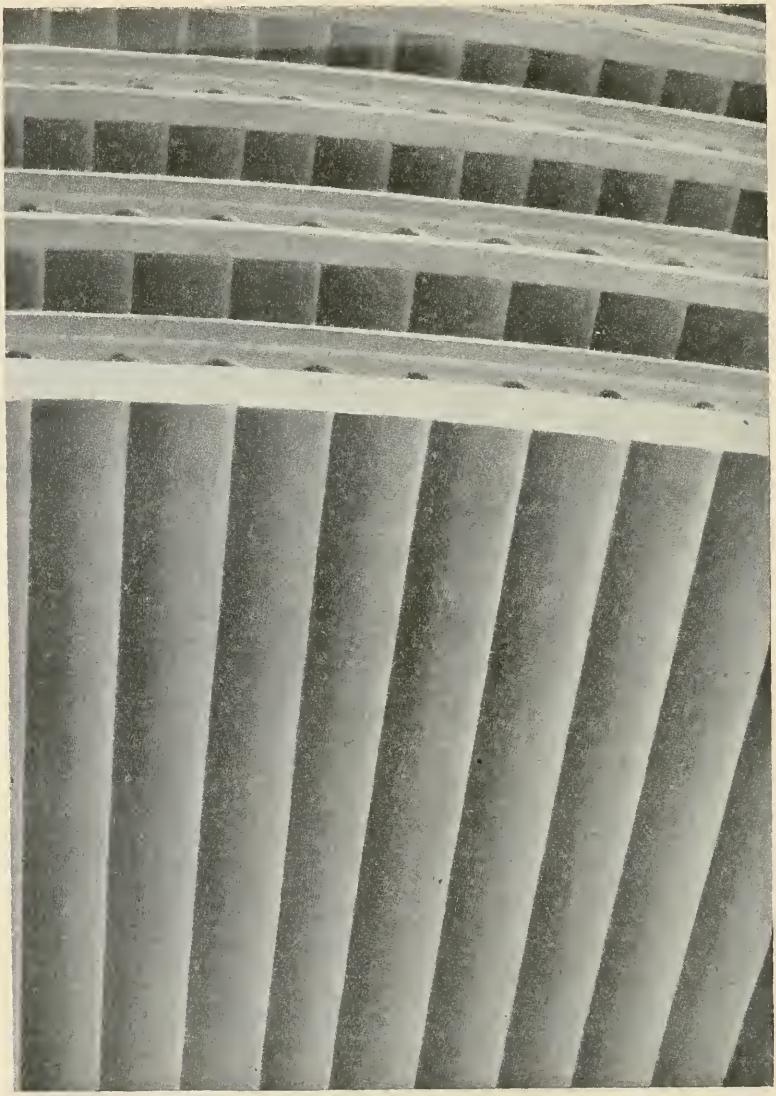


Fig. 233.—Blades and Channel-section Shrouding.

gauge, back pressure in the condenser about 1 lb. per square inch absolute. At the inlet end of the H.P. turbine the passage area between

the blades is about one-third the annulus between the rotor drum and fixed cylinder or casing; at the L.P. exhaust end of the turbine this passage area is increased by using " wing " blading set nearly or quite in line with the axis, to about 86 per cent of the annulus next the eduction pipe.

"MAURETANIA" TURBINES

	H.P. Turbine.			L.P. Turbine.		
	Gauge Pressure.	Height of Blades.	No. of Rows.	Gauge Pressure.	Height of Blades.	No. of Rows.
	lb.	in.			in.	
Receiver . .	150	$2\frac{1}{2}$	15	5 lb.	8	7
2nd expansion .	113	$3\frac{1}{8}$	15	Atmosphere	$10\frac{1}{2}$	7
3rd ,, .	87·5	4	15	4 in.	$13\frac{1}{2}$	7
4th ,, .	63	5	15	$9\frac{3}{4}$,,	18	7
5th ,, .	43	$6\frac{1}{4}$	15	$16\frac{1}{2}$,,	22	8
6th ,, .	$31\frac{1}{2}$	$7\frac{3}{4}$	15	$21\frac{1}{2}$,,	22	8
7th ,, .	$21\frac{1}{2}$	$9\frac{3}{4}$	15	24 ,,	22	8
8th ,, .	13	12	15	26 ,,	22	8

Parsons Combined "Impulse" and "Reaction" Blading.—In this system there is a single initial stage of " Impulse " blading, Figs. 234 and 235, consisting of the nozzles and three rows of fixed blading each followed by a row of rotor blading.

The grooving, Fig. 234, for the roots of the rotor blades is $\frac{3}{4}$-inch deep with seven serrations, and for the fixed blades in the cylinder $\frac{1}{2}$-inch deep with four serrations. The filling pieces between the blades are fitted in a double layer, but in the casing grooves the usual practice of a single layer is followed.

In the more recently constructed Parsons turbines, each blade is machined with its corresponding filling piece solid with it.

The radial blade clearance is about $\frac{3}{32}$-inch for both fixed and moving blades, and axially the clearance is about $\frac{1}{4}$-inch between blade rows; between the nozzles and first row of moving blades the axial clearance is about $\frac{3}{16}$-inch.

The reaction blading is of the usual Parsons type, Fig. 231.

The advantages claimed for the Combined system over the ordinary Parsons type are a shorter turbine, economy of weight and space, simplification in having only two shafts instead of three or four, and a slightly improved efficiency with a slower-running screw propeller.

The nozzles, Fig. 235, are grouped in a nozzle box A forming part of the cylinder or fixed casing. The " impulse " rotor blading is fitted on a rim B forged in one piece with the dummy drum attachment C, and

secured to the forward wheel D of the reaction turbine E. The disc carrying the "impulse" rim B has lightening holes F, and there is stated to be a uniform pressure throughout the stage G with no axial steam thrust. The cylinder or fixed guide blades (Fig. 234) extend only for an arc of the circumference corresponding to the area of the

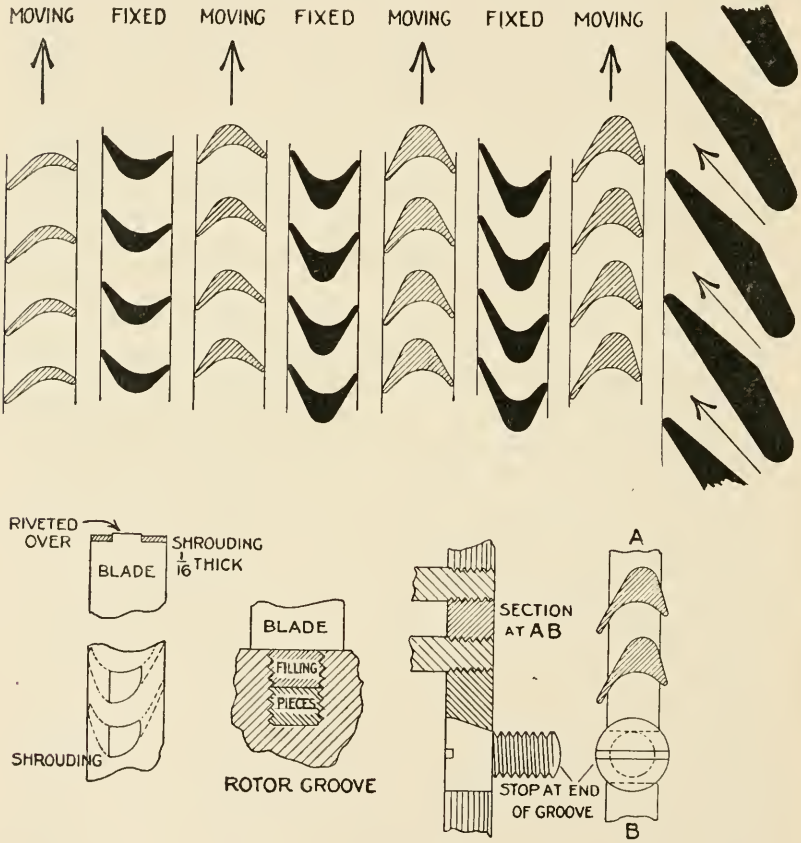


FIG. 234.—Parsons "Impulse" Blading.

respective nozzle frames. The moving blades are fitted throughout the whole circumference of the rim B, Fig. 235.

The nozzles are of the convergent-divergent type, and are designed for a pressure fall from 200 to 75 lb. per square inch, which is equivalent to nearly one-fifth the total heat drop when superheat is not used.

For *cruising and low powers*, the valve groups are shut off pro-

portionately, and by this means the pressure drop can be increased to give a fall from 200 to 10 lb. per square inch, corresponding to about two-thirds the total heat drop. At low powers, nozzles of greater divergent area are desirable, and could be substituted by special arrangements for the purpose, and the angle of inclination of the nozzle to the plane of rotating blading should be increased from 20 degrees to 30 degrees.

For " astern " working in destroyers the nozzles have the same throat area as the " ahead " nozzles, with a pressure drop from 200 to 40 lb. per square inch, corresponding to one-third the total heat drop. In cruisers, the " astern " throat area is larger than in the

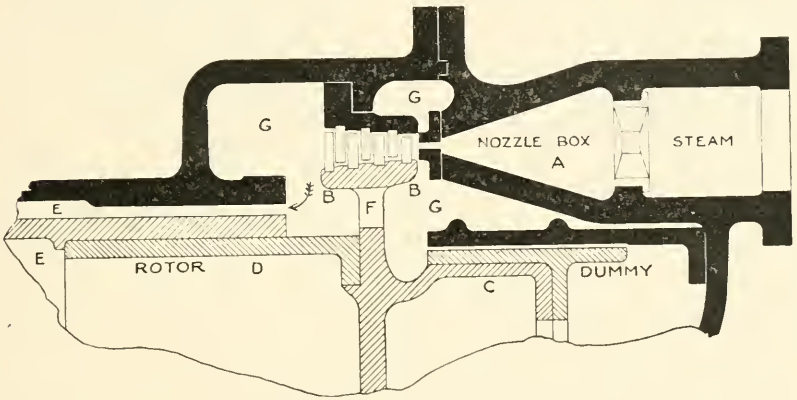


FIG. 235.—" Impulse " Part of Parsons Combined " Impulse " and " Reaction " Turbine.

" ahead " nozzles by about 25 per cent, passing the same quantity of steam with a pressure drop from 150 to 30 lb. per square inch.

The " ahead " and " astern " impulse blading is arranged throughout for a rotor velocity of 166 ft. per second at full power. As shown in Fig. 234, all the blades are shrouded, and the blade ends are riveted over the shrouding, and then machined flush with the shrouding band, which is fitted in segments each about 20 inches long, with an expansion clearance between the butts of the segments.

Ljungstrom Radial Flow Reaction Blading.—In this system there are alternate concentric rings of blades moving in opposite directions of rotation. Steam is admitted near the centre of the blade system and expands radially outwards so that the casing of the turbine comes in contact with the exhaust steam only at the condenser pressure and temperature. By fixing one rotor at a time it is possible to obtain

rotation in opposite direction, Ahead and Astern, but the shafting arrangement does not easily admit this in practice. The " action " and " reaction " stream lines are very similar to that shown in Fig. 231 for Parsons Blading.

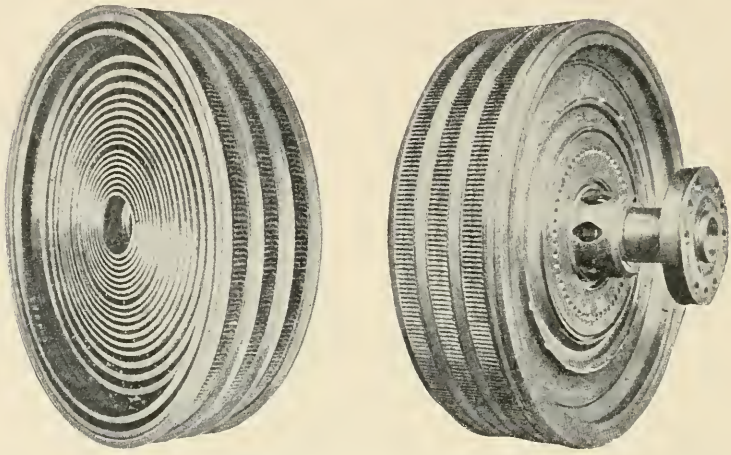


Fig. 236.—Ljungstrom Rotor Wheels.

Fig. 236 shows twin rotors which dovetail into each other, as shown in sectional elevation in Fig. 259, in which it should be noticed that the *axial* length of the blades corresponds to the shape of the nozzle, first converging and then diverging considerably towards the outer periphery

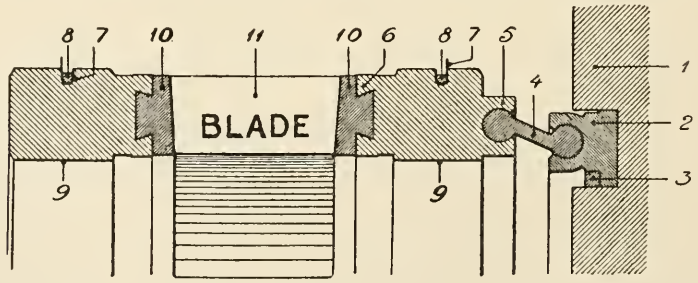


Fig. 237.—Ljungstrom Blade Ring.

and exhaust ; the increase in diameter and corresponding increase in passage area is insufficient in itself to provide for the necessary relative increase in volume of the steam from 1 to about 200.

Fig. 237 shows a section of a single blade ring, attached to its disc 1 ; 2 is a seating ring, and 3 its corresponding caulking strip ; 4 is a dumb-

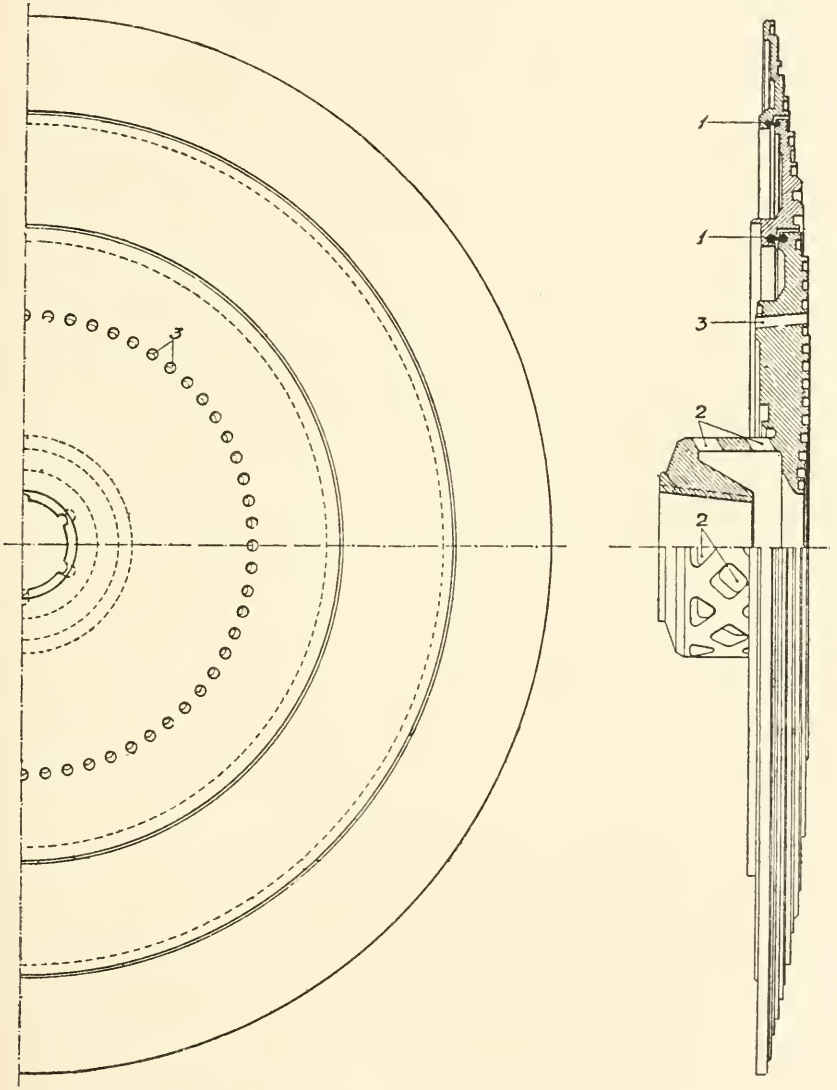


Fig. 238.—Ljungstrom Turbine Rotor Disc.

bell sectional expansion ring which carries the strengthening rings 9 and 9 between which the blades 11 are fitted. The blades 11 are dove-

tailed at the ends 10 to fit into the strengthening rings 9 and secured by a rolled-in edge at 6. The expansion ring 4 is similarly secured by rolled-in edges at 5. A radial packing ring or thin strip is fitted at 7, and secured by a caulking strip or ring 8, to check leakage from the blade region into the spaces next the discs.

The blades are accurately machined from steel bars and welded into the discs 10, which are then machined down to the correct profile as shown in the figures. The expansion ring 4 allows the blade ring to adjust its diameter to variations of temperature. In cross section the blades correspond closely to that shown in Fig. 234, for those near the centre to that shown in Fig. 231, for those near the exhaust, where in addition to the support given by the main rotating discs one or two intermediate rings are fitted, Fig. 236.

Fig. 238 shows a half view and section of one of the rotating discs, which is built up in sections connected by expansion rings 1. One face of the disc is grooved to receive the blade seating ring and caulking strip. The other, and outer, side is arranged to receive the labyrinth packing described in the next chapter, and has also an extension forming a hub. The hub is provided with a number of openings 2 through which the steam is admitted to the blade system. From the nature of the arrangement each rotor is overhung and supported at the axle by one bearing and the alternator or driving shaft. The rotor shaft, which lies within the turbine casing and is subject to heating from contact with the steam, is made hollow to allow for the fluctuations of temperature and simultaneous expansion or contraction, and thus avoid relative movement. The passages 3 through the disc are provided to allow for admission of steam at an intermediate stage of the blading system for overload conditions of working.

CHAPTER XXI

TURBINE CONSTRUCTION—DETAILS AND ADJUSTMENT

Reversing.—A little study of the form of the Parsons reaction blading (Fig. 231) makes it evident that the turbine is not reversible by changing over the steam and exhaust, and consequently it is necessary to use a separate turbine for reversing the direction of rotation of the shaft, unless gearing (which would be impossible for any but the lowest powers) is used for the purpose. For marine propulsion, the power and speed required for astern working can be less than that used for going ahead, and consequently the astern turbine is smaller, and also less economical, than the ahead turbine. In many cases it has been found convenient to install the astern turbine within the exhaust casing (as shown in Fig. 239) of the L.P. turbine. In this position the astern turbine L revolves with the ahead turbine K and in the vacuum created by the condenser and air pump; in consequence of this low pressure the astern turbine offers a very low resistance when revolved by the working of the ahead turbine. In a similar manner, when the astern turbine is at work, the ahead turbine revolves in a medium of low resistance, viz. the vacuum created by the condenser and air pump. A valve and pipe are fitted for connecting the steam ends of the astern and ahead turbines respectively for ahead and astern working, with the exhaust space.

Referring to Fig. 239, K is the L.P. ahead turbine of a series of turbines, and steam enters at A from another turbine. After passing through successive rings of fixed and moving blades, B, C, D, E, etc., the steam enters the condenser through the exhaust space H. For entering and leaving harbour, etc., the steam inlet A is closed by a self-closing valve, and high-pressure steam can be admitted directly from the boiler either through the pipe P *for ahead working* or through O *for astern working*. The blades of the astern turbine are fitted with reference to the axis in an opposite manner to that

in which the ahead blading is fitted, so as to produce rotation in the opposite direction.

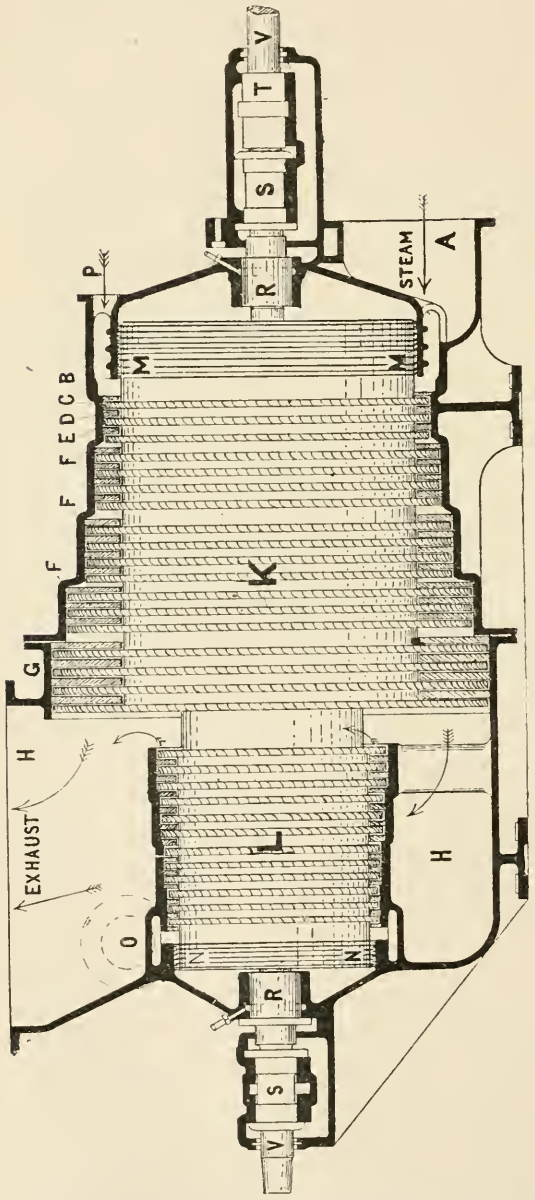


Fig. 239.—Astern Turbine and L.P. Ahead Turbine.

The moving blades of both ahead and astern turbines are mounted on the hollow drum *LK*, which is carried by shaft *V* and bearings *S* near the ends of the fixed casing. Steam packed glands are fitted at *R*, and a thrust block, usually at the forward end of the ahead turbine, at *T*. To prevent the passage of high-pressure steam into the exhaust space, in the interior of the drum, dummy packing rings are fitted at *M* and *N*, and by this means axial balancing is partly effected.

In some instances, with large powers, the Astern turbine is entirely separate from the Ahead turbine, but is fitted on the same shaft.

Closed Exhaust System for Turbines.—In many vessels the exhaust steam from the auxiliary engines, which are usually great steam con-

sumers, is used in the L.P. turbines in addition to steam from the boilers coming through the higher stage turbines. The general arrangement is similar to that fitted with reciprocating engines, but the steam is led into a particular position on the turbine where it can be best utilised and where the pressure from the other turbines is about the same or slightly less than that in the closed exhaust system. In some small vessels the closed exhaust system is capable of driving the ship at 8 to 10 knots and is therefore useful for cruising or keeping steerage way.

The system is also used for warming through the turbines while raising steam in newly lighted boilers, and in several vessels the exhaust steam is used for warming the feed water in either contact or surface heaters.

Balancing Axial Thrust.—In all axial-flow turbines which embody reaction, such as the Parsons type particularly, there is an

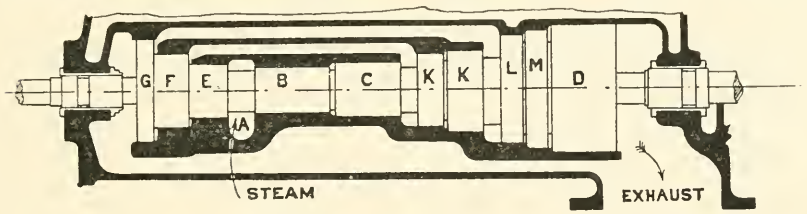


FIG. 240.—Balancing Axial Thrust.

axial thrust on the shaft in the same direction as the flow of the steam. This thrust is due primarily to statical differences of pressure on the moving blades between the inlet and exhaust. Movement of the rotor and blades, within the fixed cylinder, in an axial direction is prevented by a thrust block and collars on the shaft, which are described later. No work is done when no motion is produced, and, consequently, the axial thrust does not occasion expenditure of energy except in overcoming the friction in the thrust block caused by the difference of pressure. In Parsons turbines this statical thrust is balanced by *dummy pistons* fitted on, and incorporated with, the rotor drum, so that as little pressure is exerted on the thrust collars as possible, and thus thrust friction is almost entirely eliminated.

A common method of balancing the axial thrust of turbines, other than those used for marine propulsion, is shown in Fig. 240 (Westinghouse-Parsons type). The steam enters at *A*. Blading is fitted in connection with the various parts, *B*, *C*, *K*, *K*, *L*, *M*, *D*, of the rotor,

and the difference of inlet and exhaust pressure tends to push the rotor to the right, or exhaust, end. Dummy pistons *E*, *F*, and *G* are fitted to the left of the inlet *A*, and by suitable channels in the casing the difference of pressure on these dummies tends to push the rotor to the left and thus balance the axial thrust on the blades. The diameters and areas of these dummy pistons are made suitable to the various total pressures corresponding to the various over-all diameters of the successive enlargements of the rotor and increased length of blades.

Vibration Balancing.—The rotating parts of a turbine tend to set up vibration, if they are not symmetrically arranged about the axis of rotation. Great care is taken in the manufacture by machining the drums both inside and outside, as well as the shaft and other parts, but even with these precautionary measures there is still some tendency to cause vibration. After the rotor is completed and mounted on its shaft, it is usually tested by rolling on two perfectly horizontal knife-edged supports under the bearing positions. Any bias tending to retain the rotor in any particular vertical plane is corrected by removing or adding weight to compensate the bias. The above is termed *static balancing*, and deals only with forces, not couples.

In all cases *dynamic balancing* is also necessary, especially in modern turbines running at high velocities. In this case, the turbine is erected and run under steam in the shop at a speed somewhat greater than that anticipated at full power on board. Tests are made by means of standard fittings or a vibrometer, of which there are many varieties, but little is gained by their use. In practice, corrections are mostly experimental, and weight is added or removed according to experience. The process is sometimes tedious and prolonged, but good balancing always conduces to better working quite apart from overcoming the inconvenience caused on board by indifferent balancing. Dynamic balancing deals with couples, as well as forces, and may reveal distortion under the effects of temperature and dynamic load.

In some instances the outer hollow shafts have been the cause of excessive vibration owing to the internal bore being out of truth with the external surface; in one case the bore was eccentric as much as $\frac{1}{2}$-inch, and the shaft itself was plugged with wood and a crescent-shaped section of steel plate to obtain a fairly true balance statically. After this treatment vibration was considerably decreased; this was a war expedient, but of course the shafts were weaker than the design called for and should not be accepted as good practice in peace time

in which probably the shafts would be condemned and others of true bore fitted.

Dummy Piston Packing.—The dummy piston, in addition to absorbing part of the axial thrust, is used to prevent the passage of steam from the inlet to the exhaust without acting on the blades. Grooves, as shown in Fig. 241, are cut in the outer circumference of the dummy (or part of the rotor equivalent to it) at *M* (Fig. 239). These grooves are made with a slightly projecting lip, and a corresponding number of inwardly projecting rings are fitted in the fixed casing or cylinder so as to enter the spaces formed by the grooves. By careful adjustment of the axial position of the rotor within the fixed casing, using the thrust block for this purpose, the fixed rings barely touch the lips of the grooves, and leave a very small space through

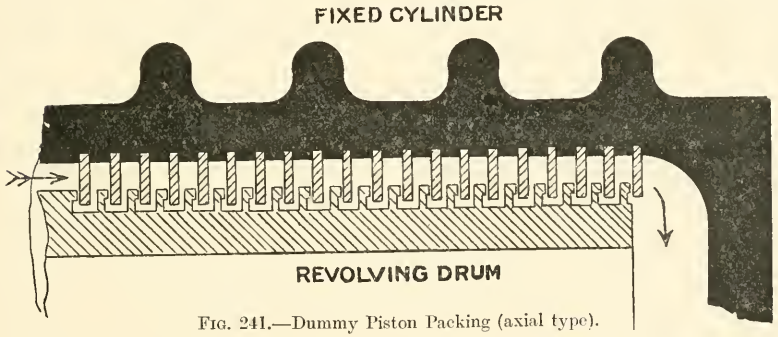


FIG. 241.—Dummy Piston Packing (axial type).

which the steam can leak. The centrifugal action of the rotor collars between the grooves assists in preventing leakage, and the escaping steam expanding into each *labyrinth* space in turn through a number of such expansions, twenty or more in practice, the total leakage is thus extremely small. The frictional resistance of the dummy packing to the rotation of the drum is almost inappreciable, because there is no actual metallic contact of the lips and rings. For the same reason internal lubrication is unnecessary. The method shown in Fig. 241 is generally adopted for the ahead inlet end of the turbine and nearest the thrust block.

For the astern turbine end, allowance has to be made for difference in expansion of the material of the casing and rotor, and the axial type of labyrinth packing is not susceptible of sufficiently close adjustment. A radial type of packing is therefore generally adopted, as shown in Fig. 242, which, however, embodies the same general principle. Rings

of thin gun metal (now usually V-shaped at the proximate edge) are fitted so as to project outwardly from the rotor and inwardly from the casing, so that they nearly touch the opposite fixed or moving parts respectively. Plenty of room is allowed axially between each pair of fixed and moving rings, so that there is no possibility of metallic contact.

It has already been stated that movement lengthways is prevented at the forward end by the thrust block, and therefore expansion must be allowed for in the turbine casing towards the after end. This is obtained by elongating the holes in the casing standards, in an axial direction, for the holding-down bolts. The axial clearance between the fixed and moving blades is much larger than the clearance between the dummy piston rings, and thus prevents the blades from coming in contact with each other.

Shaft Packing Glands.—The glands *R*, in Figs. 239 and 244, at

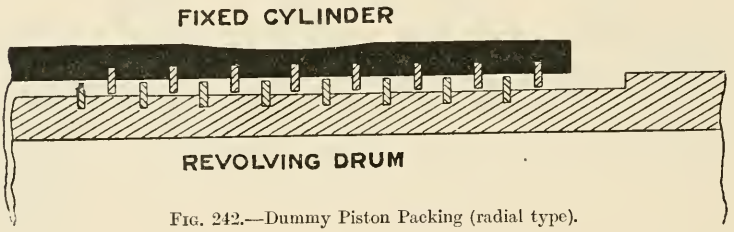


FIG. 242.—Dummy Piston Packing (radial type).

the parts of the shaft where it leaves the casing, are steam packed. In Fig. 244, a number of split Ramsbottom rings fit loosely into grooves between collars on the shaft inside the gland, and are free to move along the casing and to expand, and they are usually allowed to revolve with the shaft. This gland is not used in any way as a bearing, and only serves to prevent the admission of air into the exhaust space by admitting a very slight quantity of steam or water, which does not reduce the vacuum. The steam or water, either low-pressure steam or hot-well water, is admitted through a small pipe in connection with the grooves, and situated about one-third the length of the gland from the outer end. In cases where the internal exhaust pressure is greater than atmospheric pressure, such as the H.P. turbine, the steam pipe is connected through a two-way cock with the condenser. An alteration in power, producing a variation in the exhaust pressure, necessitates a slight readjustment of the steam pressure in the packing gland; a slight external leakage, which causes no inconvenience, is generally

allowed at all powers. No oil is admitted to these glands for lubrication, and as there are no internal bearings or rubbing surfaces in the turbine, internal lubrication is unnecessary.

A ring, W, is shrunk on to the shaft, to throw off any oil that may work along the shaft from the bearing S, and it thus prevents oil working into the packing gland and exhaust steam.

In some vessels a modified type of spindle gland packing is used, as shown in Fig. 243. The part next the exhaust space is fitted with V-shaped rings on the labyrinth principle, used for the dummy radial packing, and the outer part is fitted on the Ramsbottom ring principle described above. Steam of moderate pressure is admitted to the inlet

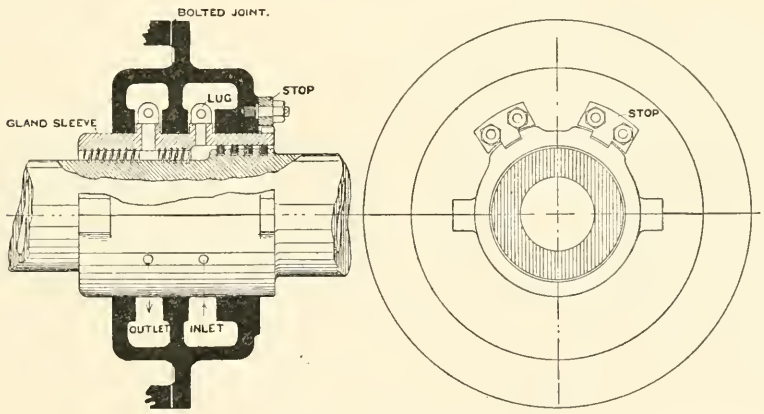


FIG. 243.—Shaft Packing Gland.

space and enters the space round the shaft through holes drilled in the gland sleeve, and finds its way out through the outlet at a reduced pressure through holes similarly drilled in the sleeve between the inner and middle sets of packing. The gland sleeve is contained within a casing secured by a bolted joint to the turbine casing. The sleeve casing is made in two semicircular parts, and when disconnected the two parts can be moved clear of the sleeve. The sleeve is also made in two semicircular parts and can be removed from the spindle for examination of the packing. The whole arrangement as described thus allows the gland packing to be examined or renewed without lifting the upper half of the turbine casing. A slight axial movement of about $\frac{20}{1000}$ths of an inch is allowed for the gland sleeve, which is prevented from revolving by the stops shown in the figure.

Shaft Bearings.—Only two bearings (*S* in Figs. 239 and 244) are necessary for the turbine itself, one at each end of the case. The bearings are of white metal, contained in cast steel or brass, and are supported on adjustable pieces fitting into the engine framing. The white metal fits the shaft accurately over nearly the whole bearing area ; there is little clearance at the butts compared with a reciprocating engine bearing. Oil is *forced* into the top half of the bearing by a pump, through holes *U* in Fig. 244, near its mid-length, and works out at the ends into an oil well ; an oil groove is cut along the top of the bearing, leading from the inlet holes to nearly the ends of the white-metal lining.

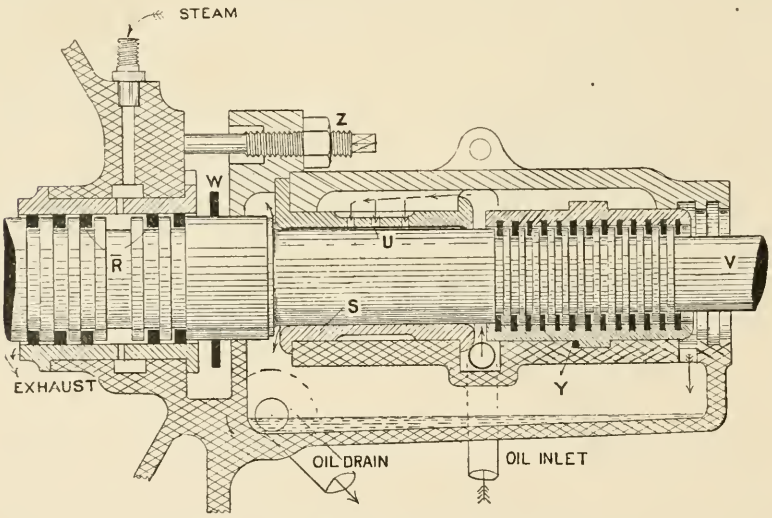


Fig. 244.—Shaft Bearing and Thrust Block (old type).

The oil gravitates through a drain pipe into a tank fitted with a cooling coil, and, after being filtered, is returned back through the bearings by the pump. The thrust bearing is lubricated in the same way, which, as the bearings run for some years without any appreciable wear, is most successful.

Thrust Bearings and Adjustment.—It has already been stated that the difference of pressure of the steam promotes an end thrust along the axis of the shaft. When a direct drive marine screw propelling turbine is turning ahead, this end thrust is nearly balanced by the thrust of the propeller ; assuming, of course, that the steam travels along the drum from forward to aft. Part of the difference is balanced

by a *dummy piston* (*M* in Fig. 239) formed at the forward end of the drum. Any other slight difference is absorbed by an external thrust bearing.

The *thrust bearing*, *T* in Fig. 239, is usually fitted at the forward end of the main propelling turbine, just clear of the casing. The bearing is made in two halves, upper and lower, the former of which is adjustable along the shaft by studs, *Z* in Fig. 244, with check nuts and butting

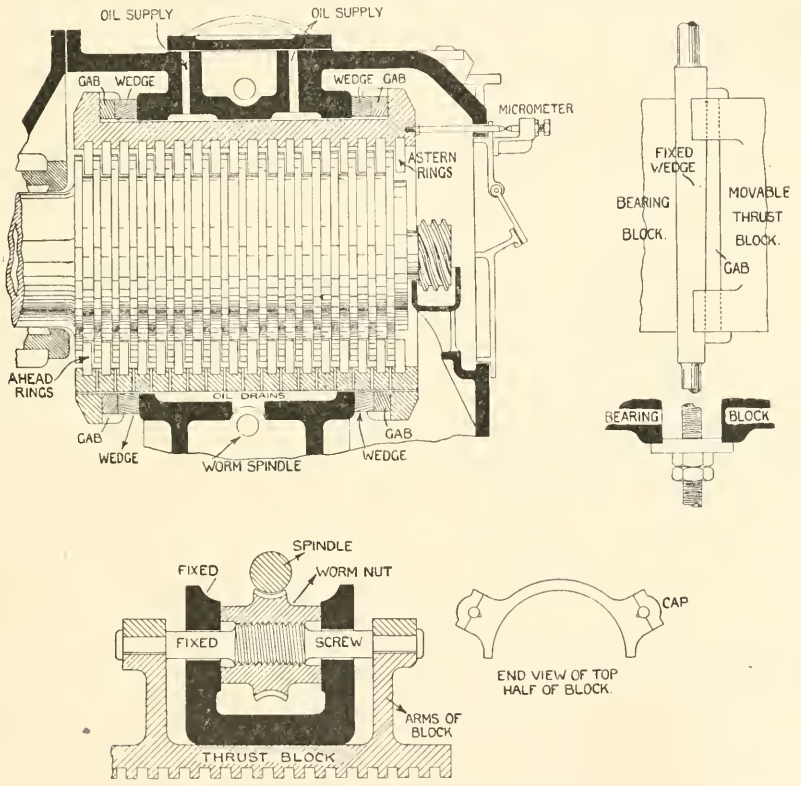


FIG. 245.—Thrust Block Adjustment Gear.

on the casing. The collars on the shaft bear on separate half-rings, which are fixed in the upper and lower parts of the bearing, and fit loosely between the collars.

The *end play* of the drum is adjusted by drawing the shaft forward until the dummy piston (*M* in Fig. 239) bears hard up against its collars; then a ring is fitted at the after end *Y* (Fig. 244) of the collar on the lower part of the thrust block, of sufficient thickness to set back the

dummy the required amount of clearance. The end clearance of the dummy packing rings varies from 0·003 to 0·025 of an inch, according to the type and size of turbine. The upper half of the thrust block is then adjusted, to prevent the shaft working aft, and to take any difference of end thrust, when the rotation of the propeller is reversed for going astern.

Some large-powered units are fitted with the type of thrust block shown in Fig. 245, in which the upper half of the thrust bush is secured and adjusted without removing the cap. Worm-geared nuts are carried

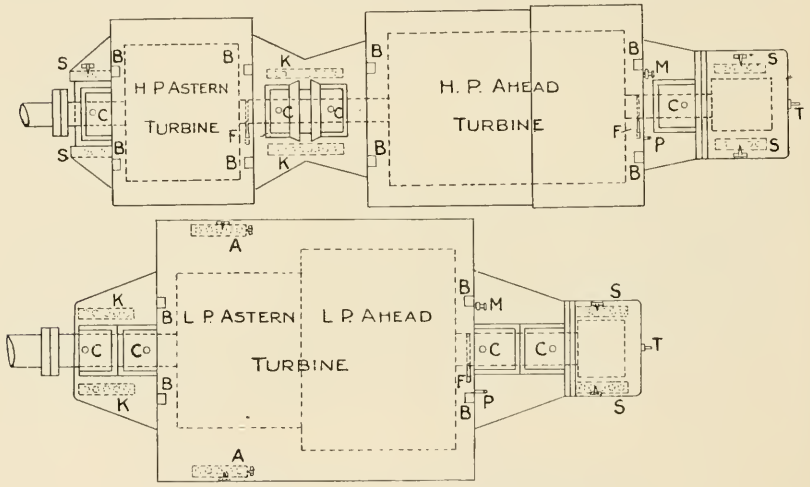


Fig. 246.—Plan of Turbine Adjustment Gauges.

A. Sliding foot gauges, axial and athwartships. *S*. Sliding foot gauge, axial only. *K*. Fixed feet. *B*. Facings for bridge gauge. *C*. Micrometer wear-down gauges. *M*. Dummy micrometer gauge. *P*. Poker gauge. *F*. Finger gauge. *T*. Thrust (oil) clearance micrometer gauge.

by the thrust block casting, and the traversing screws are carried by brackets projecting from the half bushes. Traverse gab and wedge pieces secure the bush in axial position relative to the outer casing.

The adjustment of this type of block is made by first slackening both top and bottom wedge systems. The rotor is then drawn forward, by using the traversing screws, until the dummy rings are in contact, and a reading is then taken by the dummy (micrometer) gauge. The lower bush is then set back until the thrust rings come in contact, and a reading is taken on the adjusting gauge indicator.

The upper half is next set aft to give the required dummy clearance, and readings again checked. If correct, the top wedges are secured

and the adjusting gear is used to bring the lower bush aft until the shaft collars are in contact; if the readings are correct the lower wedges are then secured. The upper half bush is next set to give the requisite axial **Oil** clearance of 0·005 to 0·015 inch, which can be checked by the oil clearance micrometer.

The Michell single collar thrust is now usually fitted for taking the turbine thrust of geared turbines; the arrangement is shown in Fig. 267.

Parsons Turbine Adjustment Gauges.—In all turbine installations, temporary *cold* adjustments must be made to allow for differences of expansion caused by changes of temperature, and subsequently the permanent adjustments are made when the turbines are thoroughly warmed up and working under normal conditions of full power. A plan of the gauges used for making these adjustments is shown in Fig. 246 for a pair of a four-shaft set of Parsons marine turbines. In other turbine arrangements similar gauges are used where they are necessary.

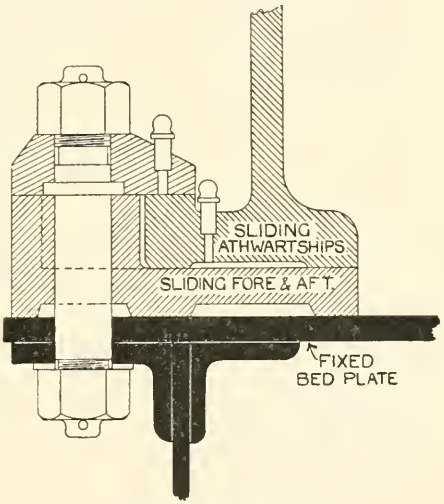


FIG. 247.—Detail of Sliding Foot.

The two principal allowances to be made are:

(1) For the expansion of the turbine casing independently of the seating which forms part of the ship's structure;

(2) For the expansion of the rotor and spindle independently of the casing.

The thrust block seating is usually fitted at the forward end of the turbine casing to which it is rigidly attached, and the feet must therefore be able to move axially in the same direction as the casing when affected by change of temperature. At the opposite, or after, end of the turbine the holes in the feet for the holding-down bolts must be elongated for alterations in the turbine length or axial expansion. For large turbines a similar allowance is made for athwartship expansion at the feet. The detail of a sliding foot is shown in Fig. 247, and a plan of a sliding foot gauge, as sometimes fitted, is shown in Fig. 248.

The gauge (Fig. 248) consists simply of dial and pointer operated by pivotal pins for which the spring eliminates any errors from backlash. Where no gauge is fitted, direct measurement is very frequently used in conjunction with standard marks on the fixed and sliding parts,

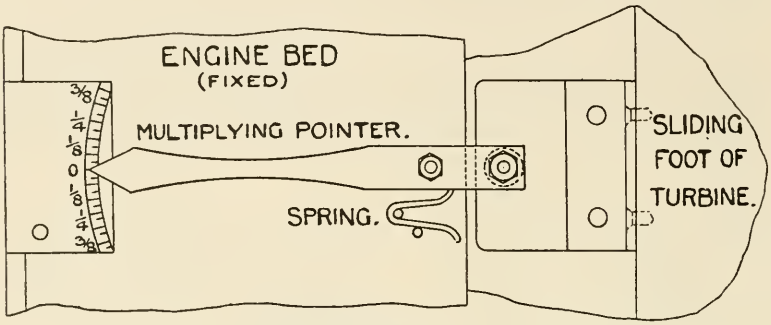


Fig. 248.—Sliding Foot Gauge.

and until the expansion attains its original trial conditions, it is considered dangerous to work the turbine.

If a Cruising turbine is fitted, it is situated at the forward end of the main turbine, and the after feet of the main turbine being fixed, the others must be fitted as sliding feet. Any difference of expansion

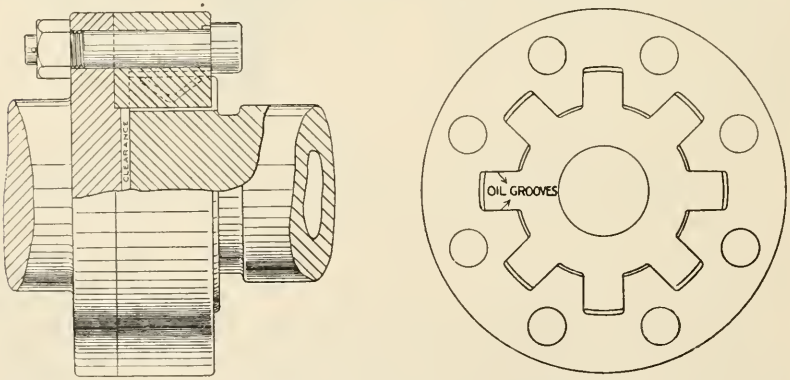


Fig. 249.—Turbine Shaft Expansion Coupling.

between the two turbine rotors and shafts independently of the casings is allowed for by fitting a sliding or expansion coupling, as shown in Fig. 249, between the ends of the adjacent rotor spindles.

A Poker Gauge (Fig. 250) is used when the turbine is *cold*, to verify the clearance of the dummy packing rings. The gauges are fitted

symmetrically on opposite sides of the top centre line at the forward ends of the H.P. and L.P. turbines.

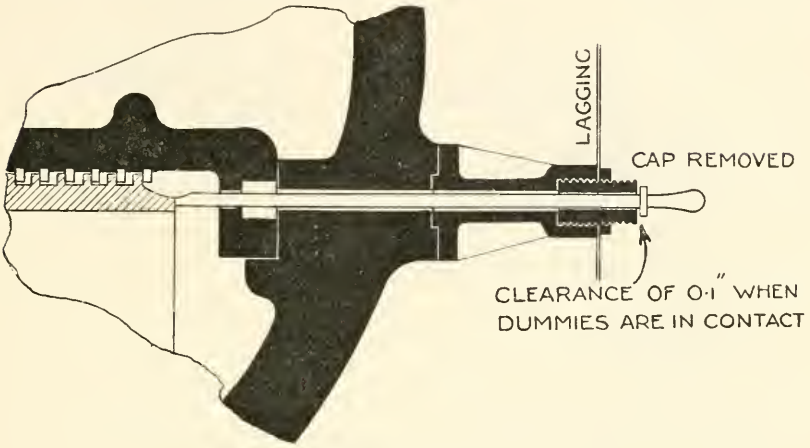


FIG. 250.—Poker Gauge.

The Dummy Micrometer Gauge (Fig. 251) is also used for the same purpose as the Poker gauge.

A Wedge Gauge (Fig. 252) is used when the turbine is *cold*, to test the clearance allowed between the turbine bearing brass and the shoulder of the rotor shaft. This clearance is usually checked before and after steaming, and is usually about $\frac{1}{4}$-inch.

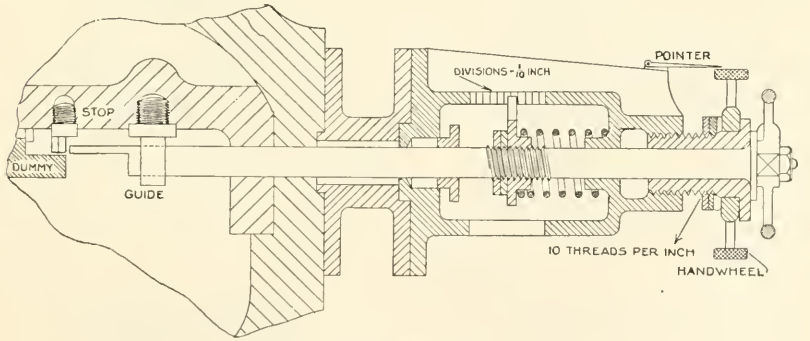


FIG. 251.—Dummy Micrometer Gauge.

A Finger Gauge (Fig. 253) is used for the H.P. *astern* turbines, and is usually attached to the forward end of the cylinder casing opposite the V-groove on the rotor shaft. The gauge is originally

set in the central position opposite the groove, and the movement of the rotor must not then exceed $\frac{1}{4}$-inch in either direction.

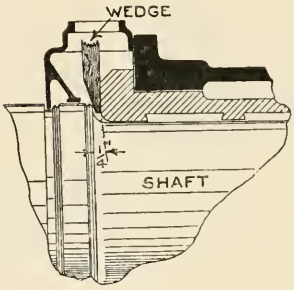


Fig. 252.
Bearing Clearance (Wedge) Gauge.

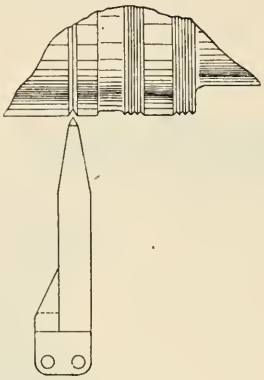


Fig. 253.
H.P. Astern Finger Gauge.

A Finger Gauge (Fig. 254) is also used for the *ahead*, or thrust block end of the turbine.

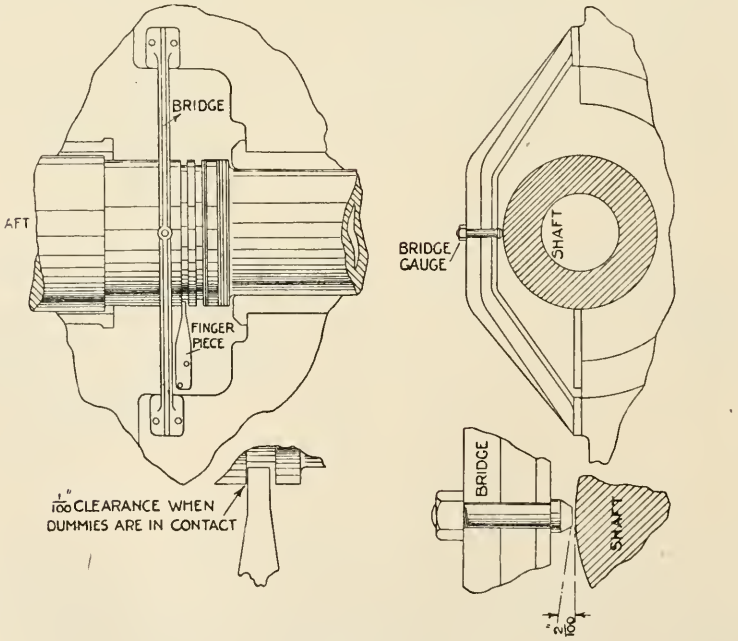


Fig. 254.—Finger Piece and Bridge Gauge.

A Bridge Gauge is also shown in Fig. 254 for measuring the wear of the main bearings after removing the cap. For this purpose a **Micrometer Gauge** (Fig. 255) is also used and more frequently fitted ; this gauge can be inserted through the main bearing cap when the turbine is running.

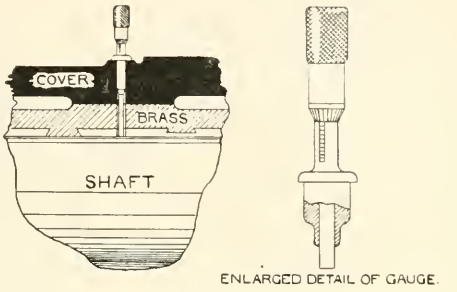


Fig. 255.—Wear-down Micrometer Gauge.

Turbine Lifting Gear.—Fig. 256 shows one of the L.P. turbines of the *Mauretania* open for inspection by lifting the upper half of the turbine casing by means of a 30 B.H.P. electric motor. The guide columns *A* are 7 inches in diameter, and are bolted to the turbine casing and braced at the top or otherwise secured to the deck beams. The two columns at each end of the turbine are embraced by guide brackets on the casings. Forged steel columns *B* can be inserted between the upper and lower casings when requisite to lift the rotor. For lifting the rotor a cast-steel strap lined with white metal is slung round the shaft between the main bearing and the gland, and the slings are then connected with the lifting gear and the rotor lifted and guided by the guide columns. Means are also provided for lifting and transporting various parts, such as the caps of the main bearings, parts of the eduction pipe, covers of thrust blocks, etc., and the lower parts of the main bearings can also be removed.

The essential purpose of the guide columns is the protection of the blading, and this system of guidance is common with all turbine installations. The different makers vary the arrangements, but it is desirable to make a standard system suitable for all turbine plants, and during the war standardised sets of lifting gear would have been a considerable asset in saving time and transport of these essential fittings.

Forced Lubrication.—A duplicate system of forced lubrication is now generally fitted to all turbine installations and is shown diagrammatically in Fig. 257. The larger ships have also a separate system for plummer blocks and shaft bearings.

The pump draws its supply of oil from an oil tank through a strainer and delivers it at 30 to 40 lb. pressure into a filter, and thence through

a cooler to the bearings, where a delivery pressure of about 10 lb.

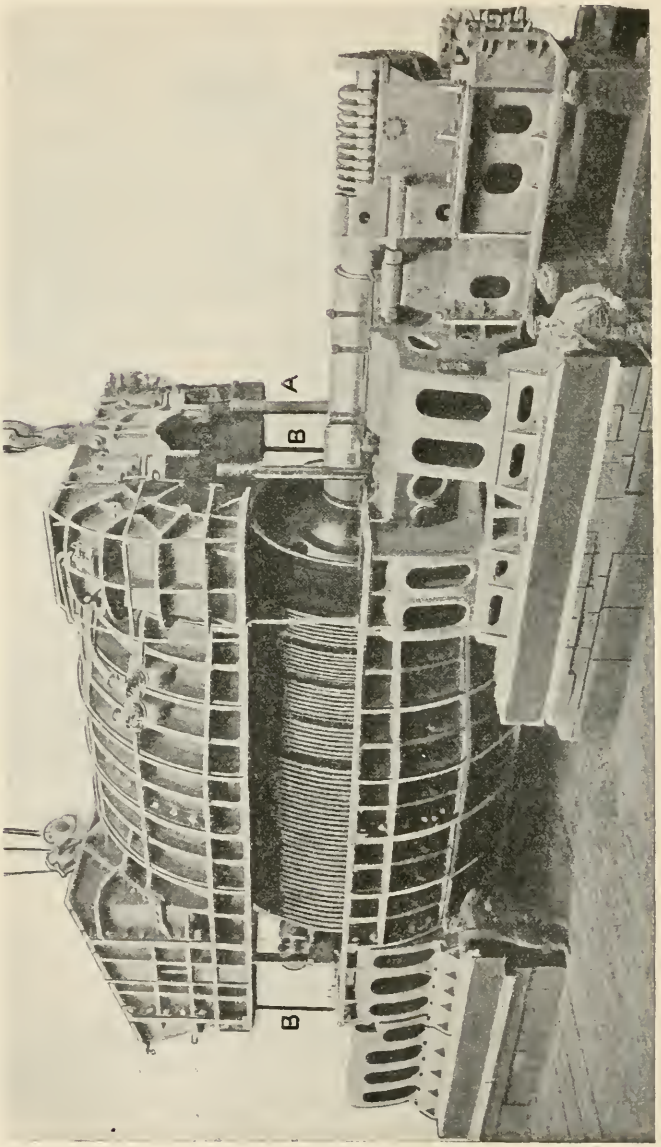


Fig. 256.—Turbine Lifting Gear.

is considered normal. From the bearings the oil gravitates into

the spaces below and drains back to the tank through a return pipe.

A reserve oil tank is fitted as high as possible in the engine room so that gravity feed can be used in case of necessity, as when cleaning the tanks, etc. A settling tank is also sometimes fitted with steam coils for promoting circulation to separate the oil from water. All the tanks are fitted with oil level gauges, and generally sounding rods and plug holes are supplied for checking the gauges. A hand pump is fitted to the drain tank for removing water or oil as required.

With a well-fitted system of forced lubrication the leakage is very

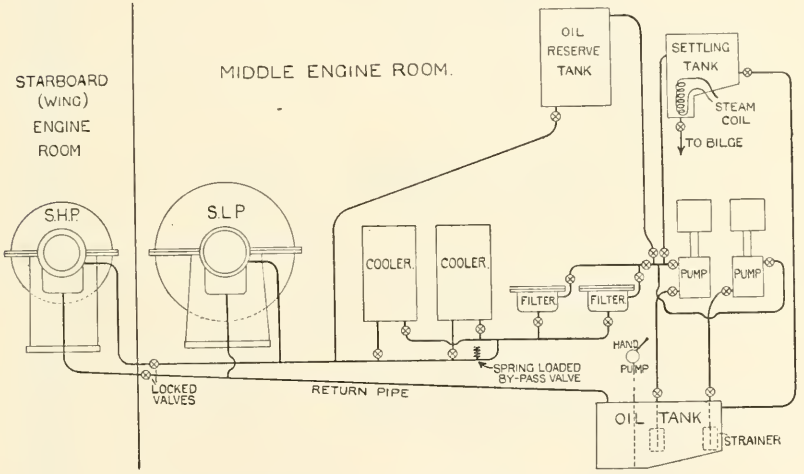


FIG. 257.—Diagram of Forced Lubrication System.

slight and any loss of more than 2 gallons per 24 hours should be considered abnormal and promptly investigated. Too much care cannot be taken to prevent waste and gritty matter from getting into the system, because choking or grinding must cause disaster if it is not checked at once.

Fig. 258 shows the Parsons system of cutting the oilways in the lower half of a main bearing. A thermometer, test cock, sight door and a pressure gauge are usually fitted for use when working.

Oil grooves are generally turned on the spindle near the bearings to throw off oil and to prevent creeping, and wipers and baffles are also fitted as shown in Fig. 252 and other figures.

Fig. 435 shows another system of cutting the oil grooves, but the Michell system is likely to supersede all other systems in the near future for large and medium-sized bearings (Fig. 178).

Ljungstrom Radial-flow Turbine.—This turbine is of the outward radial-flow reaction type. An enlarged cross section of one of these turbines is shown in Fig. 259 in which the steam fills the interior of the casing *d*, and thence enters the blading system from the central position of the hollow hubs *W*. The turbine discs and the blading were described and illustrated in the previous chapter, and are here shown in cross section with their corresponding blading interposed, one above the other in alternate rings. At the exhaust end there are two rings of blades for each turbine supported by two intermediate rings or discs in addition to the two turbine discs proper; this arrangement gives the necessary opening passage for the much-expanded exhaust at about this position and at the same time the requisite support for very long blading.

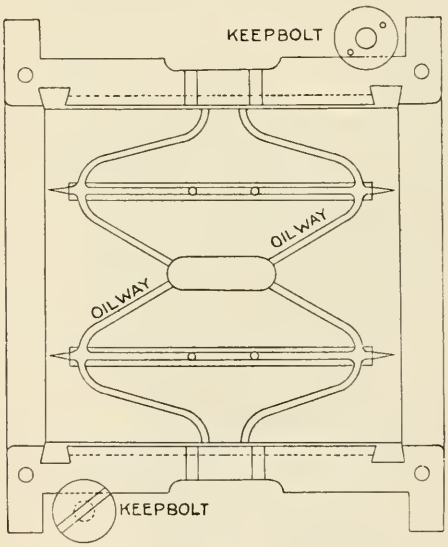


Fig. 258.—Plan of Bottom Half of Main Bearing.

On each outer side of the rotor discs are discs *C*, which are grooved and fitted with labyrinth packing running within corresponding packing on similar discs which are fixed and attached to the casing *D*. The arrangement of this packing is shown by the inset section, in which *A* and *B* are the fixed and moving discs respectively. Constriction of passage is obtained by caulking in nickel strips at the tips of the projections between the grooves and then bending them over to cover the space in the groove as closely as practicable. The total number of constrictions is thus equal to the sum of the grooves in the two discs.

Advantage is taken in this arrangement to obtain axial balance of the two rotors by proportioning them so that the axial pressure of steam, which varies from the initial pressure down to the exhaust

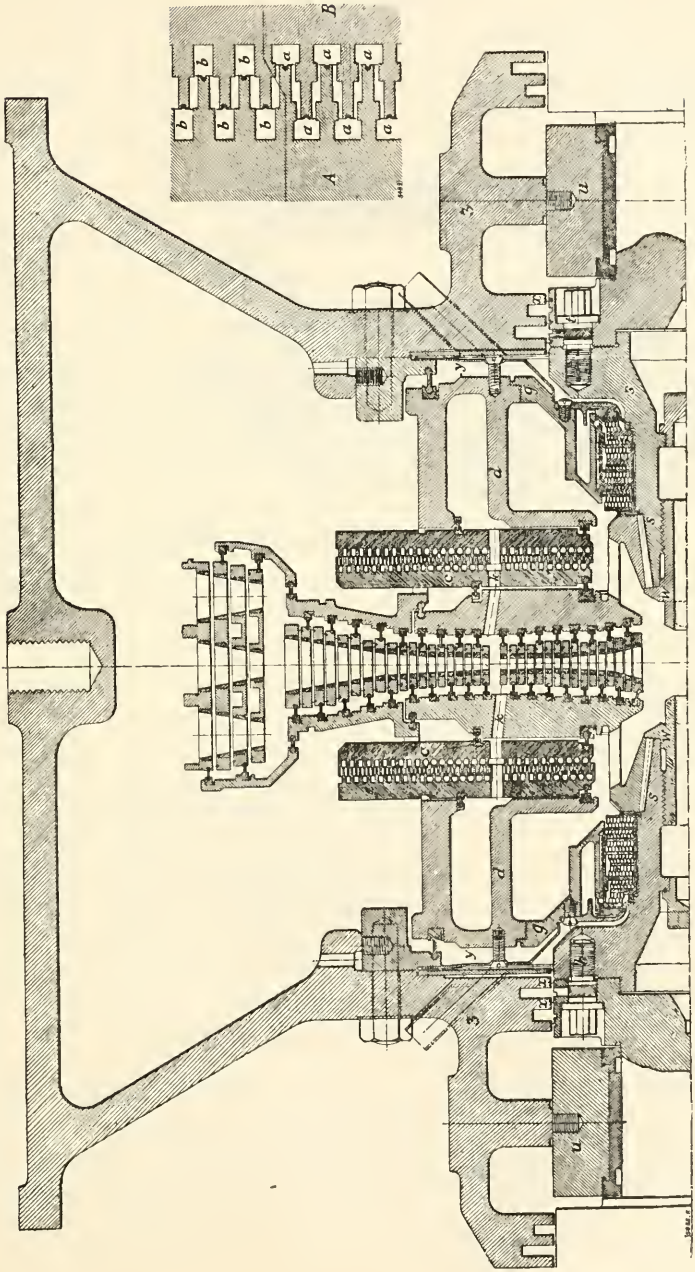


FIG. 259.—Sectional Elevation of Ljungstrom Turbine (Upper Half).

pressure, and tends to force them apart, is balanced by the labyrinth discs.

The shaft packing is shown in Fig. 260 in section for the right-hand side of Fig. 259 where it is also shown. Any steam leaking through the glands has to pass through the numerous constrictions shown. The leakage is carried away by a pipe to a special condenser, and it is claimed that with a pressure of 180 lb. per square inch it does not exceed 150 lb. per hour. This small percentage of the total steam consumption is obtained by assembling the gland with practically no clearances and allowing it to create its own clearances by wear in the first instance; the fine ends of the fins simply rub away and their mass is too small to cause distortion of the gland by the heat generated in the process.

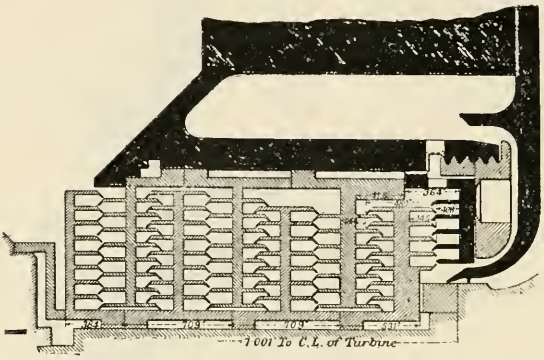


FIG. 260.—Ljungstrom Shaft Packing.

If the turbine is intended to take an overload, steam can be admitted to the casing space and thence through *Kk* to the blading. In this case no overload is embodied in the design, but steam at about 40 lb. pressure can be drawn off at this point through the passages *Kk* for feed heating purposes. To reduce radiation and convection losses the exposed outer surfaces of the casing *Dd* are all jacketed, as indicated at *Yy* (Fig. 259).

In the **shop** tests of these turbines the steam consumption averaged 12·535 lb. per Kw.-hour, or about 9·354 lb. per B.H.P.-hour; but in the **ship** the efficiencies of transmission have also to be considered.

De Laval Turbine.—Fig. 261 shows a sectional elevation and a plan (with cover removed) of a 20-H.P. turbine with a single rotor wheel driving a belt pulley *M*, through gearing *HJ*. This turbine has recently been adapted for marine propulsion and is described below.

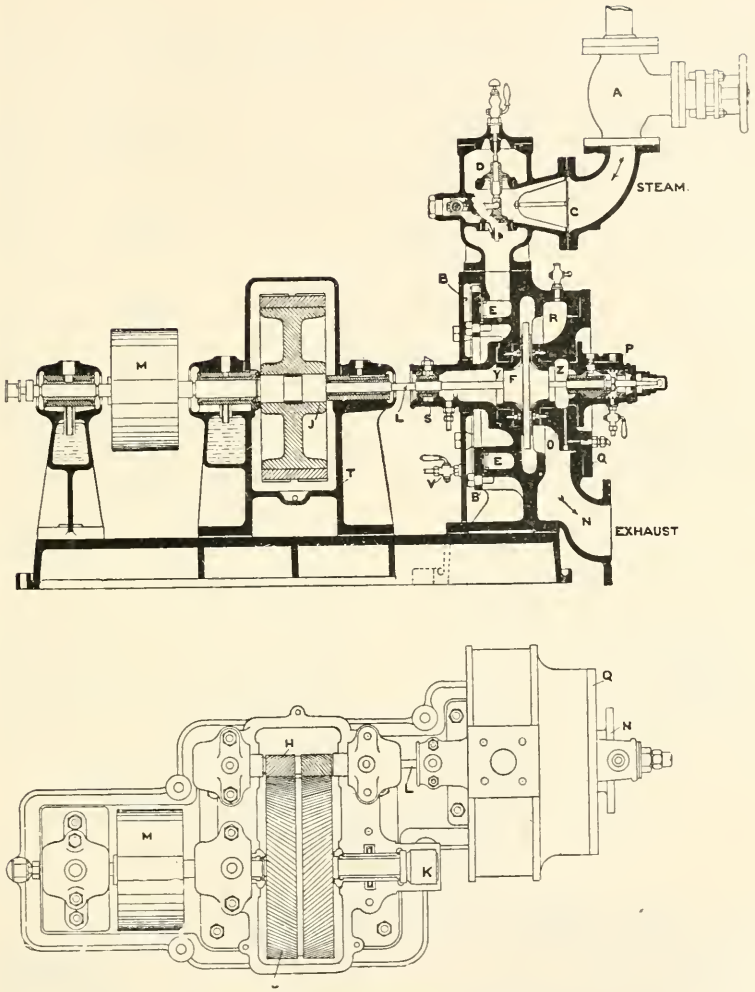


FIG. 261.—De Laval Single-wheel Turbine.

A, Steam regulating valve.
B, Face-plate enclosing circular steam space, *E*.
C, Gauze-wire filter.
D, Governor-valve box and valves.
E, Annular steam space (*C* in Fig. 222).
F, Turbine wheel.
H, Pinion wheel on turbine shaft.
J, Gear wheel on driven shaft.
K, End bearing of gear-wheel shaft (enclosing governor).
L, Turbine shaft.
M, Pulley on gear-wheel shaft (for belt driving).
N, Exhaust.
O, *Q*, End casing, enclosing wheel and exhaust space.
P. *S*, Turbine shaft bearing and casing.
R, Exhaust space.
T, Gear-wheel casing.
V, Drain to steam space.
Y, *Z*, Casings for turbine wheel.

The following table gives some of the principal sizes of De Laval turbines constructed by the English De Laval Company:

Brake Horse-Power.	Middle Diameter of Turbine Wheels.	Revolutions of Wheel per Minute.	Velocity of Blades.	Revolutions of driven Shaft per Minute.	Weight (about).
B.H.P.	Inches.	R.P.M.	Feet per Second.		Cwts.
5	4	30,000	515	3000	$3\frac{1}{4}$
15	6	24,000	617	2400	$5\frac{1}{2}$
30	$8\frac{7}{8}$	20,000	774	2000	11
		With two pulleys:			
50	$11\frac{3}{4}$	15,000	846	1500	29
100	$19\frac{3}{4}$	10,500	1115	1050	72
300	30	7,500	1378	750	$161\frac{1}{2}$

The general construction of the turbine may be gathered from Figs. 221 and 222, and the reference to Fig. 261.

De Laval Marine Turbine.—(Taken partly from *Engineering*,

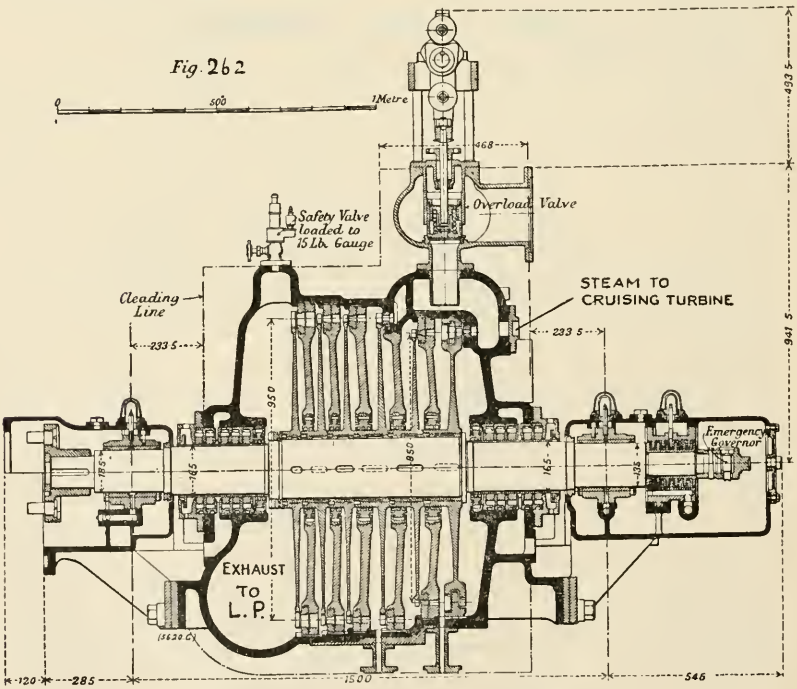
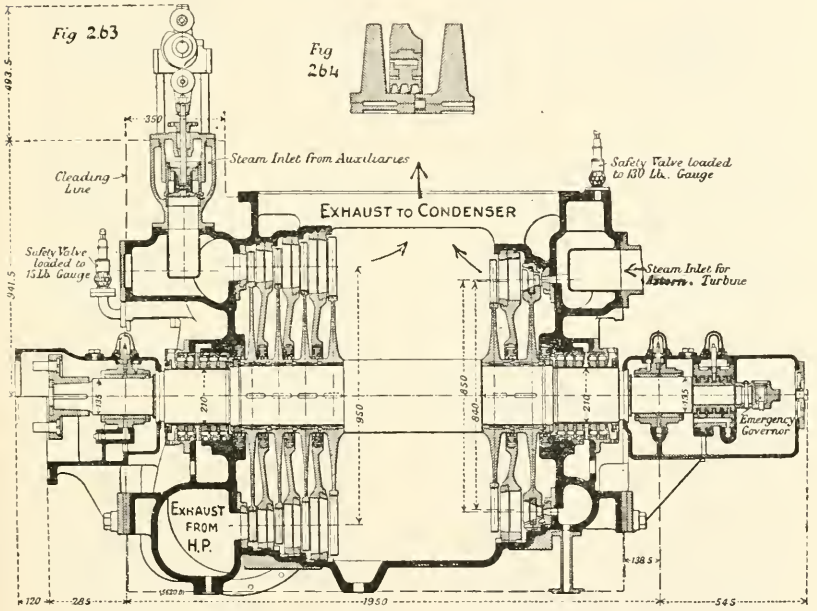


Fig. 262.—De Laval Marine H.P. Turbine.

October 10, 1919.) Two sets of these turbines have been fitted success-

fully in the Swedish destroyers *Wrangel* and *Wachtmeister*. The starboard set is fitted forward and the port set aft in separate engine rooms divided by a bulkhead. Each set consists of one H.P. and one L.P. turbine on separate shafts geared into 8 to 1 single reduction gearing of the usual double helical reduction gear type. The H.P. casing (Fig. 262) accommodates a cruising element, and the L.P. turbine casing an Astern turbine (Fig. 263).

When cruising, up to about 16 knots, steam is admitted to the



FIGS. 263 and 264.—De Laval Marine L.P. and Astern Turbine.

cruising turbine consisting of one velocity compounded wheel of two rings, and one single impulse wheel; the steam next passes through an arc of guide blades at the bottom of the casing into the first of four wheels of the main H.P. turbine. The first main guide wheel has only partial admission, but the other three wheels have all-round admission.

At full power the cruising steam valve is shut off, and steam is admitted direct to the main H.P. turbine through guide blades near the top of the casing. The cruising wheels then revolve in a small supply of steam provided to keep the temperature down, and to lessen the risk of distortion from frictional work on a restricted quantity of

steam which naturally gains admission to the cruising chamber through the first main guide blades.

In the L.P. turbine (Fig. 263) provision is made for admitting steam from the auxiliary (closed) exhaust system, but when cruising, the exhaust steam is admitted to the H.P. turbine through a branch pipe not shown in the figure.

The Astern turbine, on the right of Fig. 263, consists of one velocity compounded wheel with two rows of rotor blading followed by a single impulse wheel. It is designed for 259 revolutions per minute when developing about 40 per cent of the full ahead power. When the Astern turbine is at work, the auxiliary exhaust is deflected direct to the condenser.

The blading throughout is of the usual De Laval type, securely held and made of Monel metal. At the full power (450 revolutions per minute) the rotor blades move with a peripheral velocity of 590 feet (180 metres) per second.

The glands are fitted with spring-mounted carbon packings. The diaphragm packings are all of the labyrinth type as shown in Fig. 264 on an enlarged scale. The ring carrying the three fin packing rings is in section held together by springs and thus capable of yielding slightly in case of whipping of the shaft. Shoulders prevent the rings closing in on the shaft beyond a certain safe limit.

A trigger type of emergency valve governor and a multi-collar thrust are fitted to each turbine shaft at the forward end. The propeller thrust is taken by a bearing of the Michell type, designed for a load of 427 lb. per square inch. At the works independent tests were made in which nearly eight times this pressure was carried without trouble, a notable achievement.

Curtis Marine Turbine.—Generally speaking, the marine system consists of two turbines working separate shafts, as in the ordinary twin-screw vessel with piston engines, and reversing is effected by fitting an astern turbine at the exhaust end of each ahead turbine. The arrangement is thus fairly compact and convenient, and a cross-section of one of the twin turbines of the *Creole* is shown in Fig. 265. Each of the two turbines is 120 inches in diameter, with seven stages for " ahead " and two stages for " astern " working.

Referring to Fig. 265, the turbine consists of a cast-iron cylinder casing divided by diaphragms into seven ahead compartments and two astern compartments. In each compartment there is a separate wheel, with four rings of moving blades on the first and three on the

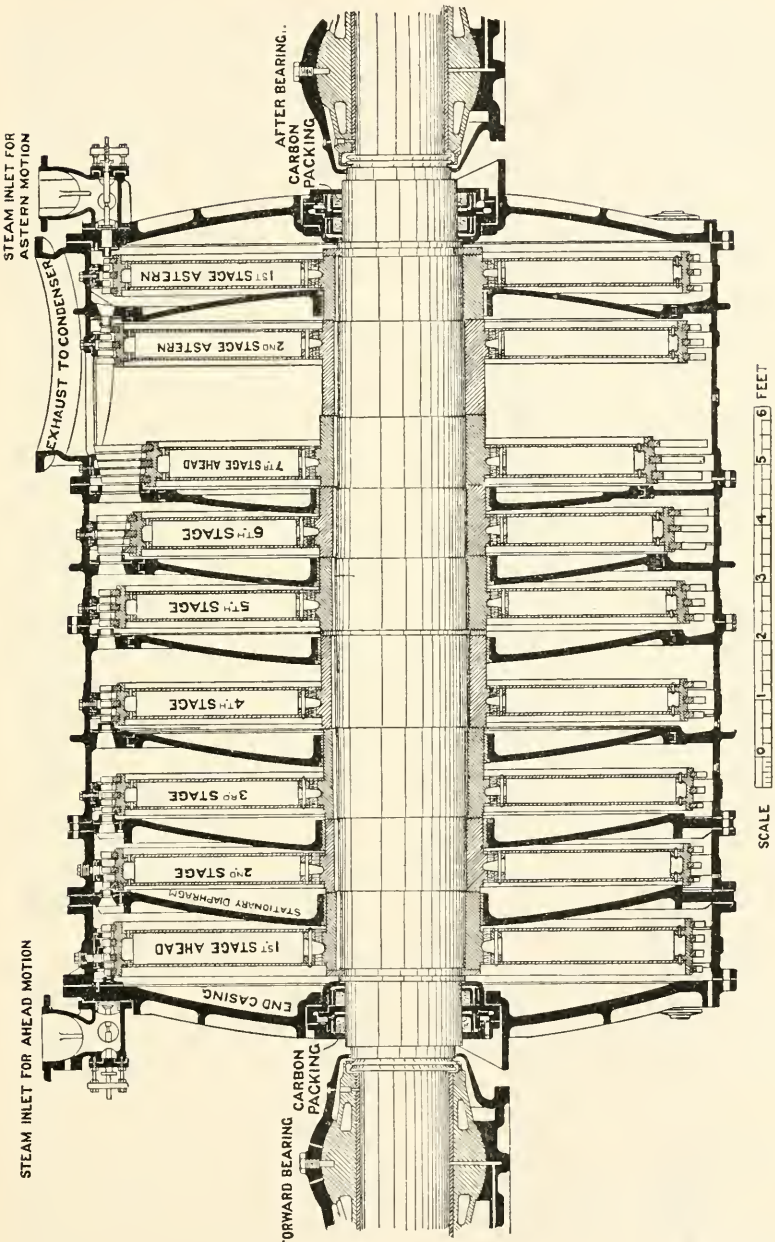


FIG. 265.—Curtis Marine Turbine (*Creole*).

remaining six of the ahead wheels. The wheels are mounted on a hollow shaft carried by self-aligning bearings. Where the shaft passes through the diaphragms, they are provided with bronze bushings having a small clearance, thus preventing appreciable leakage from one stage to another and lower one. Carbon stuffing in boxes prevents steam leaking *out* at the ahead end or air from leaking *in* at the astern end, where a vacuum exists during ahead working. The stuffing boxes are steam-packed, and are drained to the fourth-stage chamber. In the ahead steam chest there are twelve valves, each operating one of the nozzles for the first-stage wheel. For continuous running sufficient of the nozzles are opened to give the speed desired, and the ahead throttle valve is left open, thus giving full pressure in the steam chest. Nine nozzles will give full power, leaving three for overload. In the astern steam chest there are twelve nozzles, but only six are fitted with valves. For manœuvring, the nozzle valves in the steam chests are left open, and the speed is controlled by the throttle valves.

On each of the other six ahead stages two nozzle valves are fitted. Each valve closes one-fifth of the nozzle area of its stage. These valves are closed when running continuously at reduced speed, so that the proper steam pressure can be maintained in each stage. Drain pipes are fitted to connect each stage with the next lower one, so that the condensed steam in any stage can part with its heat in doing useful work. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. A common marine thrust bearing is provided for the forward end of the turbine shaft. As in the Parsons turbine, this bearing ensures the correct relative axial position of the rotor and fixed casing. Axial clearance of the blading is one-tenth inch in the first wheel, and increases up to one-quarter in the seventh. The length of the blades is increased towards the exhaust end, to allow for volumetric expansion of the steam.

In order to keep the pressure in the shell as low as possible the pressure distribution is arranged so that one-fourth of the available energy of the steam is expended in the first stage, and one-eighth in each of the other six stages. This requires the first stage nozzle to be of the expanding type, but all the other nozzles are of the parallel flow type.

Brown-Curtis Turbine.—In the type shown in Fig. 266 there are several nozzle diaphragm and wheel stages at the " ahead " high pressure end, exhausting into a drum stage with a large number of impulse bladed expansions fitted in connection with a drum. The

"astern" turbine is fitted on the same shaft, but the number of wheel and drum stages is less. In the half section of a turbine shown in Fig. 266, it will be noticed that it follows for the nozzle and wheel stages the same general form as shown in Fig. 265. The turbine spindle is subject to a very heavy stress, and is continuous through the whole length of the turbine including the drum. The wheels fit on tapered portions of the spindle with distance pieces between them, and the whole set at each end is secured by a nut. The forward or high pressure end of the drum stage is fixed on the spindle, but the after wheel (at the exhaust end) is free to move axially on the spindle, and thus allows for difference of expansion. For the Astern portion of the turbine no allowance for expansion is considered necessary, and the wheels and drum are fixed on the spindle.

In the more recent types of Brown-Curtis turbine, the drum blading has been discontinued and the general form is more of the type shown in Fig. 265. The arrangements for cruising and utilisation of the auxiliary closed exhaust are the same in principle as those shown in illustrations of the De Laval marine turbine, Figs. 262, 263. In one particular set of turbines, of recent design, consisting of one H.P. and one

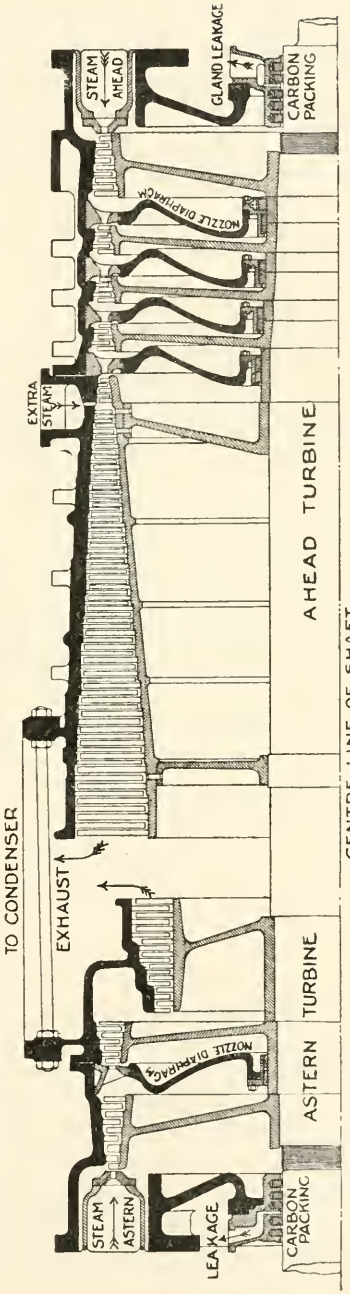


FIG. 266.—Brown-Curtis Turbine.

L.P. turbine operating on one shaft through gearing, there are in the H.P. turbine one wheel with three velocity compounded rings of blading, then two rings, then another two rings in the cruising and low power stages, followed by five single-ring rotor wheels; thus making in all eight stages, each of which is supplied by nozzles of the convergent type.

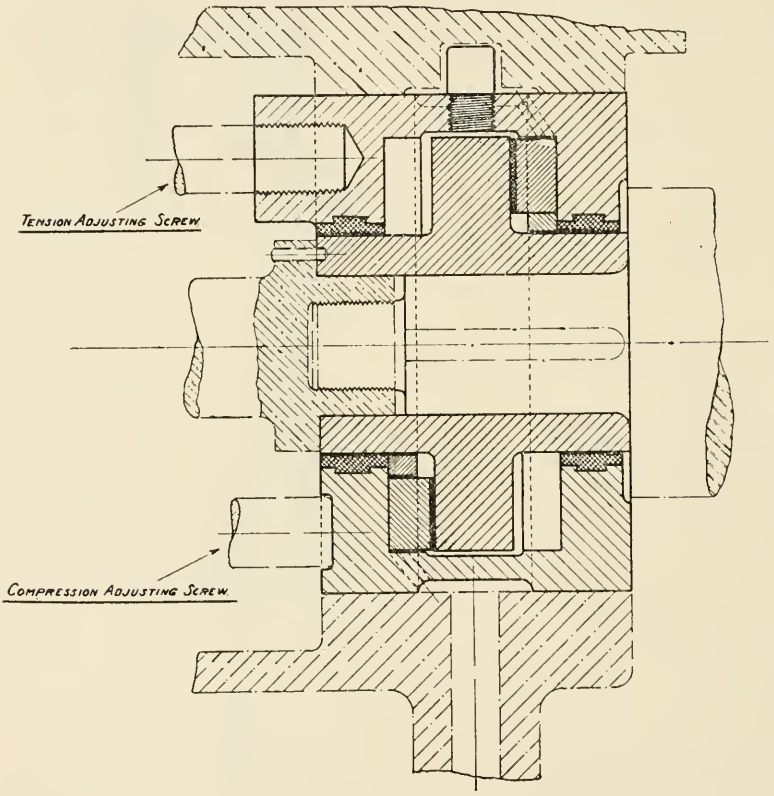


Fig. 267.—Michell Turbine Thrust Block.

In the L.P. Ahead turbine, there are eight single-ring rotor wheels with convergent nozzles for each stage; extra admission from the auxiliary closed exhaust being supplied to the second stage. The blades in this particular turbine are about 5 inches long at the inlet end, and about $11\frac{1}{2}$ inches long at the condenser exhaust end.

In the Astern turbine, which is fitted at the forward end of the L.P. Ahead casing, there is a single rotor wheel with three rings of

blades supplied at the inlet end from convergent-divergent nozzles, the steam being thus expanded to a very low pressure and the velocity created being the maximum possible for the full astern power.

Both rotor spindles are fitted with a single-collar Michell thrust block at their forward ends, of the type shown in Fig. 177, with hardened steel pivots for inclining the thrust pads. In this instance the fore and aft adjustment is of a semi-permanent nature and is effected by fitting packing rings, or distance rings, behind the spherical bearings;

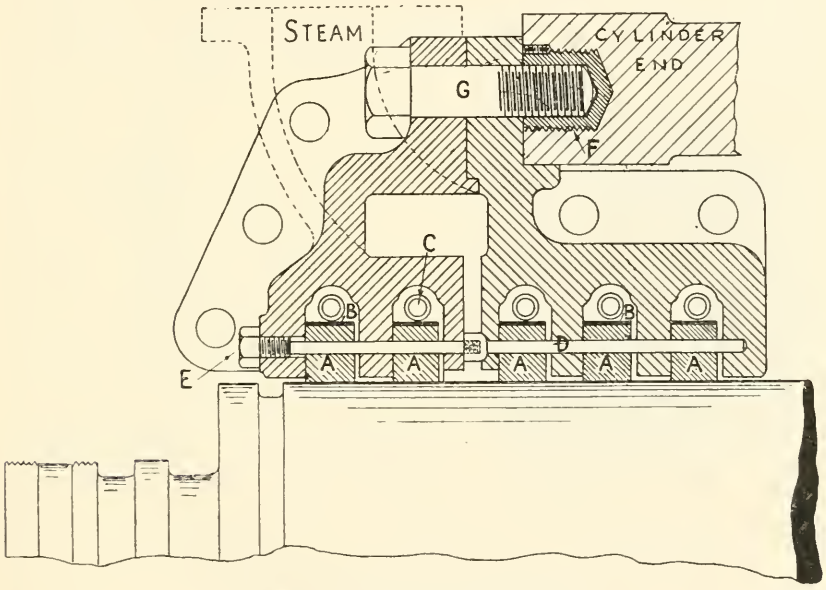


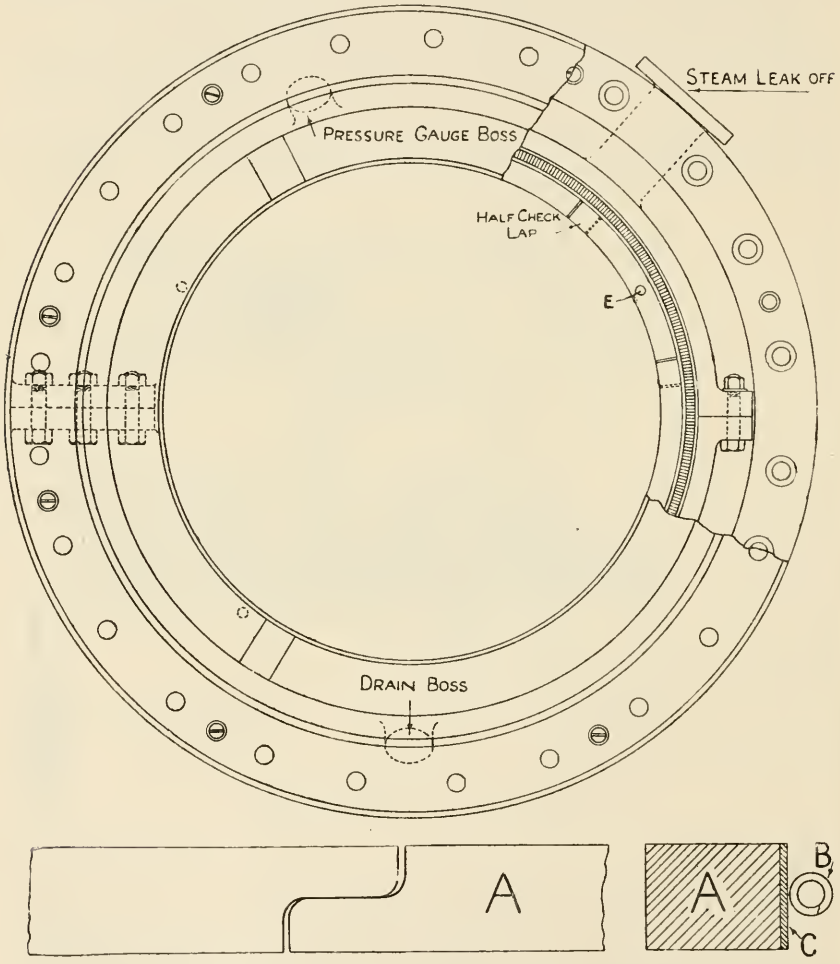
FIG. 268A.—Carbon Packing for Rotor Shafts.

this method is convenient where the axial clearance of the blading is large, as is usually the case with Impulse turbines.

In Fig. 267 another arrangement of turbine thrust block is shown as made by Messrs. Broom & Wade, and is suitable for small axial blade clearance, as in the Parsons reaction turbine. External adjustment is practicable through adjusting screws, as shown, following the method adopted in the multi-collar thrust block shown in Figs. 244 and 245.

Carbon Packing Rings.—In many turbines of the Impulse type, carbon packing is used for the shafting where it emerges into the open from the casings. The arrangement is shown in Fig. 268, which

shows three views, and consists essentially of a number of carbon rings, *A*, fitted round the spindle and kept in contact with it by a



DETAILS OF LAP JOINT FOR CARBON RING

FIGS. 268B and C.—Carbon Packing for Rotor Shafts.

circumferential spring, *B*, wrapped outside a $\frac{1}{32}$-inch thin metal cover *C*, enclosing the carbon. The carbon ring is made up of several sectors which are lap jointed and prevented from turning with the shaft by rods *D* and *E*. Leakage along the shaft past the packing is carried

away by a steam leak-off, and a drain is fitted at the bottom to carry away any condensed steam and water. The pressure is maintained, if necessary, at this position by admitting steam from elsewhere to prevent the indraught of air when the pressure at the inner or turbine casing end of the packing is below atmospheric pressure, and a pressure gauge is fitted for reference.

The gland for holding the carbon packing rings is made in two rings, each of which is composed of two semicircular parts bolted together, and the whole gland is secured to the cylinder end by screws, G, fitting into brass plugs, F, secured by set screws in the cast-iron cylinder end. Before withdrawing the outer ring, the rod, E, is screwed

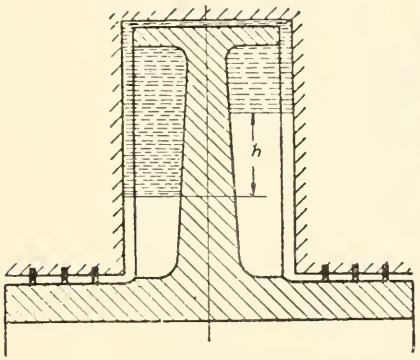


FIG. 269.
Westinghouse-Rateau L.P. Water Seal Packing.

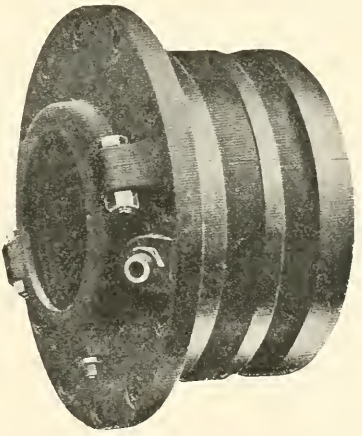


FIG. 270.—Westinghouse-Rateau H.P.
Water Seal Packing Gland.

back until it is clear of the inner rod, D, which is left in place until the inner packing is withdrawn.

Each of the nozzle diaphragms is also fitted with carbon packing in many recent types of impulse turbines, but only one ring is usually fitted for each diaphragm. In some of the older types (Fig. 265), metal packing bearings with several water grooves were fitted for each diaphragm, but in the Brown-Curtis turbine (Fig. 266), a combination of carbon and metallic bearing was fitted. The single carbon packing ring is, however, now generally fitted for each diaphragm in Brown-Curtis turbines and has been found efficient in practice.

Water Seal Packing.—In the Westinghouse-Rateau and other turbines the shafts are packed at the ends of the casing by a system of steam and water sealing shown in Figs. 269 and 270. The water

seal alone is normally used for continuous steaming, and the steam seal for stopping and starting, getting under way, etc.

The seal consists (Fig. 269) of an impeller rotating in a race in which the centrifugal force of the moving water tends to maintain an equal head in each water leg, but the difference in steam pressure and the atmosphere on the two ends of the gland forces a difference in height, h, which is allowed for in the design. Water is fed into the race from a tank about 20 feet above the gland, and in the case of the L.P. cylinder the excess escapes as steam into the turbine casing and passes away

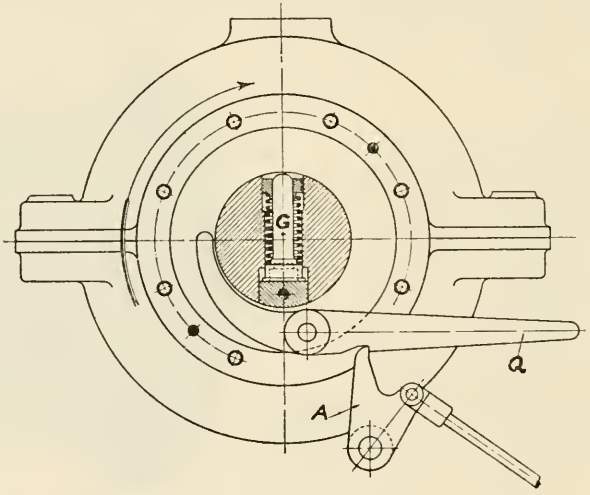


Fig. 271.—Overspeed Governor.

to the condenser. There is thus no water loss, and the water seal is completely air-tight.

When steam sealing is in use, the admission of steam to the gland is regulated so that no loss of steam or condensate occurs, and when once adjusted no further regulation is necessary because the conditions in the gland are independent of load or change in speed or pressure.

Overload Safety Governor.—The fracture of a shaft or other cause may allow a turbine to overrun the maximum speed and cause damage to the turbine itself, and therefore many turbines are fitted with an overload governor, of which a type, fitted in connection with the Westinghouse-Rateau and other turbines is shown in Fig. 271. The governor is fitted at the end of the rotor spindle, and is of the centrifugal type. The weight and its spindle or guide, G, with its

centre of gravity a small distance from the centre of rotation is opposed by a spring whose resistance is less than the increase in centrifugal force at normal speed. At abnormal speed the weight begins to move and flies out to the limit when the rounded end comes in contact with the tripper sliding catch, *A*, and thus closes the automatic valve in the steam chest (Fig. 281).

A hand lift, *Q*, or a pedal lift is also provided for emergency purposes.

CHAPTER XXII

ARRANGEMENT OF TURBINES AND TRANSMISSION OF POWER

Direct Drive. — The early turbines fitted for marine propulsion were generally of the Parsons reaction type, and to obtain a comparatively low rate of revolution of the propeller with a consequent fair efficiency of propulsion a three-shaft arrangement of turbines was used, with numerous turbines arranged to give as many stages of expansion as practicable and corresponding to the multi-stage system of reciprocating engines. Thus two or three stages were generally fitted for mercantile vessels and continuous steaming, while in warships a system of additional cruising turbines for low power steaming at corresponding low speeds was adopted.

In H.M.S. *Amethyst*, the first turbine small light cruiser, the Admiralty allotted the same weight and space as in sister vessels with reciprocating engines, and this resulted in the further adoption of turbines to the eventual exclusion of reciprocating engines in all warships of high speed and power. The introduction of transmission or reduction gearing has during the last five or six years allowed this adoption of turbines to be followed in vessels of only eight knots full speed.

Fig. 272 shows the arrangement of the direct drive of the turbines in the *Amethyst*; by *direct-drive* is meant that the turbines and propellers are connected directly by their respective shafts without the interposition of gearing or other transmission system.

For the lowest powers and speeds (Fig. 272), steam is admitted to the H.P. *Cruising* turbine *A*, and the exhaust from *A* is conducted to the M.P. *Cruising* turbine *B*, where it does more work, and then exhausts into the H.P. *Main* turbine *C*, from which the exhaust passes in two streams to the two L.P. *Main* turbines *D* and *D*, and finally escapes into the two condensers *E* and *E*. The steam is thus expanded in four stages, the two L.P. turbines forming one stage only as in four-cylinder three-stage expansion reciprocating engines.

For powers which cannot be obtained by passing all the steam through the H.P. Cruising turbine, a small increase may be obtained by admitting steam direct through the pipe *H* into the M.P. Cruising turbine *B*; or the steam can be shut off from *A* and entirely admitted through *B*. A large number of destroyers were fitted with this arrangement, and during the war, when the cruising speed was usually higher

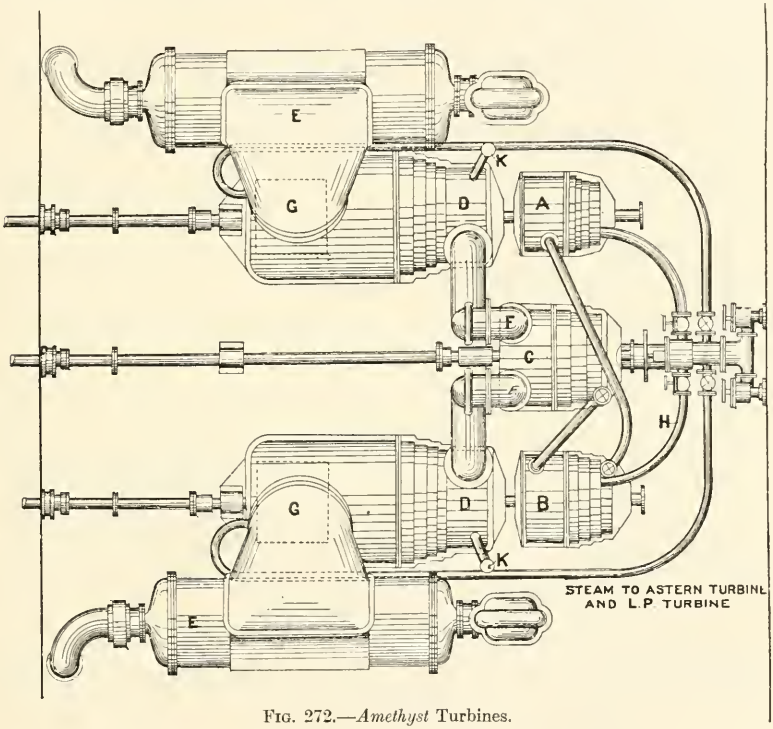


FIG. 272.—*Amethyst* Turbines.

than in peace time, the H.P. Cruising turbine was seldom used, and in some instances removed from the vessels and not replaced.

For the highest powers, the Cruising turbines are shut off and, if so fitted, disconnected; steam is then admitted directly to the H.P. *Main* turbine. If turbines are not in operation and cannot be disconnected, then the inlet and exhaust ends are connected with the condenser exhaust so that the rotors when revolving idly run *in vacuo* with no appreciable resistance. The claw-clutch arrangement for connecting and disconnecting the cruising turbines is very common in destroyers of pre-war design.

For *astern* working and backing and filling into or out of harbour, etc., an astern turbine is fitted in the same casing with each of the L.P. *Ahead* turbines, and the vessel is worked only through directly connected pipes and valves *K*, which are of the manœuvring type and are used for both ahead and astern working. Only the wing shafts are so controlled, and in this case the H.P. *Main* turbine is allowed to revolve idly with its shaft and propeller. Self-closing values *F* are fitted to prevent any back pressure returning to the H.P. Main turbine when steam is admitted directly to either the L.P. Ahead or Astern turbine.

Fig. 273 shows the three-shaft arrangement in the s.s. *Carmania*, which made a distinct link in the chain of adoption of turbines from small to large vessels. The arrangement is very similar to the *Amethyst*, but without the cruising turbines.

The *Carmania* displaces about 20,000 tons when loaded and the turbines develop 30,000 S.H.P. when running at 180 revolutions per minute at this full power. The propellers are three-bladed and one of 14 feet diameter is fitted on each shaft.

The run of the steam piping can be seen in the figure, with the various control valves. The working or regulating valves are fitted at *M*, and each set can be shut off when not required by a stop valve *S*; this valve is very necessary because the main working valves are not very close fitting, but in later vessels a better-designed valve arrangement gives good results. Self-closing valves *F* are fitted in the L.P. receivers so that when the working valves are in use the steam cannot reach the H.P. turbine.

A three-shaft arrangement which has been fitted in several instances, consists of a reciprocating three-stage expansion engine on each of the two wing shafts, exhausting into a single L.P. stage turbine on the central shaft. At the time it was a very economical arrangement, but is not likely to be repeated.

Another combination of turbines and reciprocating engines is shown in Fig. 266 of the four-shaft arrangement in the s.s. *Rochambeau* of the Compagnie Générale Transatlantique. The turbines are on the wing shafts and reciprocators on the central shafts, each set, port and starboard, being complete in itself. The exhaust from the reciprocators can be led direct into the condensers, and this system is used when working in and out of harbour, etc.; at such times the turbines are inoperative and left free to revolve with their respective propellers. No astern turbines are fitted.

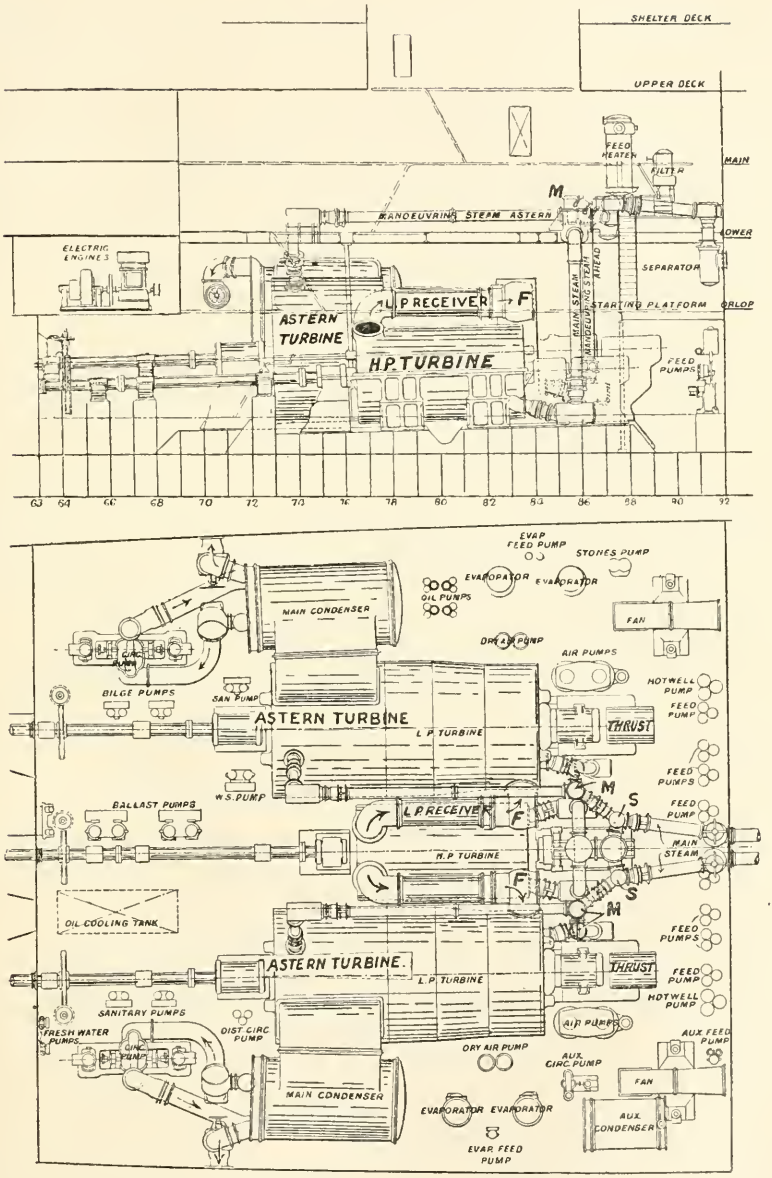


FIG. 273.—Turbine Engines of s.s. *Carmania*.

The early Dreadnoughts were fitted with four shafts (Fig. 275) in two separate engine rooms each of which is worked as a separate

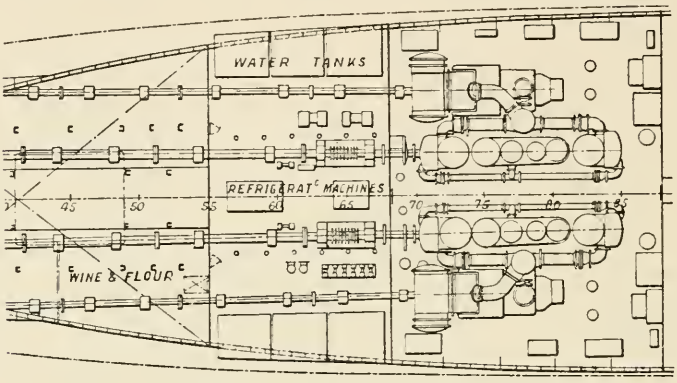


Fig. 274.—Combined Reciprocating and Turbine Engines (*Rochambeau*).

unit. At low powers running ahead the steam is used in three stages, H.P. Cruising, H.P. Main, and L.P. Main turbines; but at higher powers

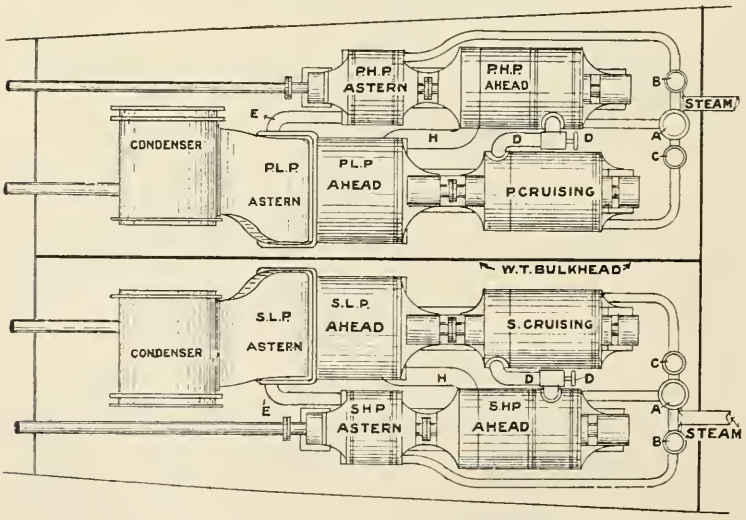


Fig. 275.—Direct-drive Turbines of H.M.S. *Dreadnought*, etc.

the H.P. Cruising turbine is not worked and its inlet and exhaust ends are connected with the condenser.

For high powers, when the cruising turbines are not in use, the

valves *C* and *D* are closed and steam admitted directly to the H.P. Main turbine through *A*, and then exhausts through *N* to the L.P. turbine.

For reversing, the valves *A*, *C*, and *D* are closed, the drains at the inlet and exhaust ends of the H.P.C., H.P., and L.P. Ahead turbines connected with the condenser, and the steam is admitted through *B* to the H.P. *Astern* turbine and exhausting through *E* to the L.P. *Astern* turbine thus reverses both shafts at the same time. The advantages of this arrangement are very considerable over many previous and some more recent arrangements; the turning moment

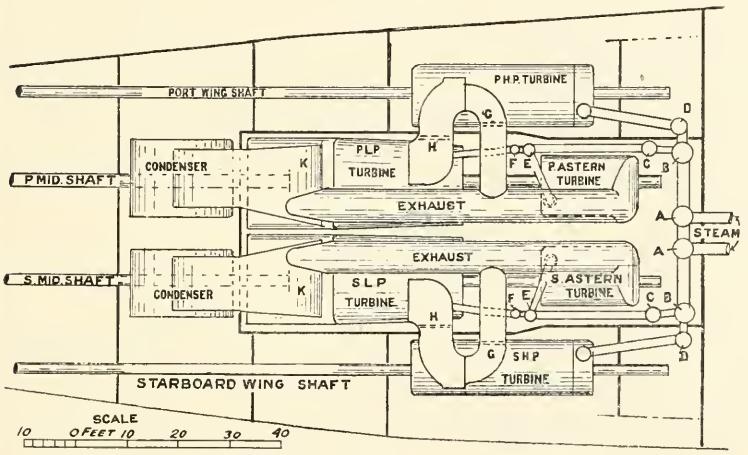


FIG. 276.—Direct-drive Turbines of s.s. *Mauretania*.

of the forces exerted by the propellers (Fig. 371) is very great and these enormous vessels are easier to manœuvre in consequence.

The four-shaft arrangement of the s.s. *Mauretania* is shown in Fig. 276; there are no cruising turbines and they are differently arranged from the *Dreadnought*.

Steam enters from the boiler rooms through two valves *A*, connected by a cross connecting pipe. The H.P. Ahead turbines are fitted in separate engine rooms on each side, and steam is admitted to them through separators, *B*, and shut-off valves, *D*. By means of sluice valves, *G* and *H*, near the exhaust end of the H.P. turbines, it is possible to work the H.P. turbines independently of the L.P. turbines by exhausting direct through the eduction pipes *K* into the condensers. The two midship engine casings each contain a separate

astern turbine and a L.P. turbine. Only the midship shafts can be driven astern, and when entering or leaving harbour, etc., the steam is shut off from the H.P. turbines by means of the valves *D*, *G*, and *H*. Steam for manœuvring is then admitted through the stop valves *C*, which are usually closed when at sea to prevent leakage, through the manœuvring valves *E* for astern working, and *F* for ahead working (L.P. only). The thrust blocks are fitted ahead of the H.P. turbines on the wing shafts, and between the astern and L.P. ahead turbines on the middle shafts. The wing propellers are nearly 80 feet *before* the middle screws, and turn inwards. The middle screw propellers turn outwards. The main condensers occupy a compartment by themselves, and other separate compartments are occupied by the auxiliary condensers and main air pumps and circulating engines. The propelling machinery and its principal accessories thus spread over nine compartments, besides several auxiliary compartments.

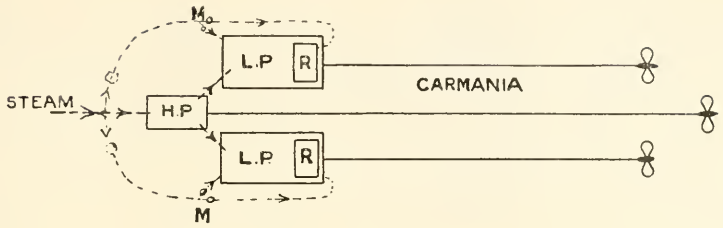
Diagrammatic arrangements of several three-shaft and four-shaft vessels are shown in Figs. 277 and 278, and may be useful for reference to direct-drive turbines.

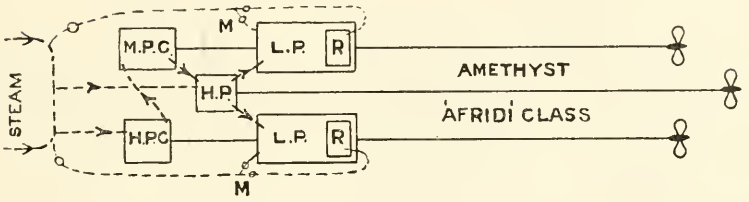
The largest direct-drive turbines built in the world are fitted in H.M.S. *Repulse*, developing about 120,000 S.H.P. on trial. The Brown-Curtis turbines are fitted on a four-shaft arrangement: two on each side of the ship in each engine room with a mid-line bulkhead between the two engine rooms. The system is thus similar in principle to the *Dreadnought*, of which the *Repulse* is a modern, but much faster type, and the two shafts on each side are operated as a single unit for manœuvring and continuous steaming.

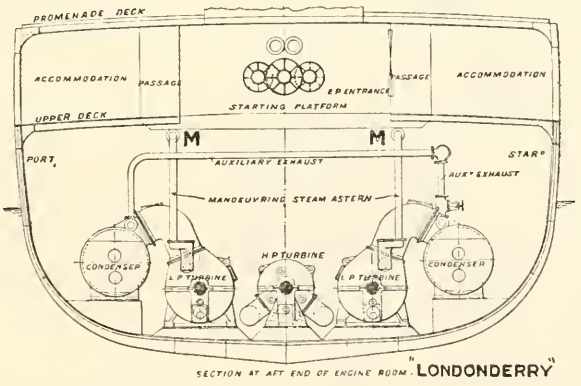
In place of the H.P. Cruising turbine in the *Dreadnought* there is a High Pressure Main turbine which is used at all powers when steaming ahead. This is at the fore end of the central shaft and exhausts into the intermediate (M.P.) turbine for ahead working on the wing shaft; from the M.P. turbine the steam is led back to the central shaft position and enters the L.P. turbine which is situated abaft the H.P. Ahead turbine and having a shaft common to the two turbines, H.P. and L.P.

Inside the casing of the M.P. turbine at its after end is the H.P. *Astern* turbine on the wing shaft; the H.P. Astern turbine exhausts into the L.P. Astern turbine fitted on the central shaft and incorporated with the L.P. Ahead turbine in the same casing.

The system is thus practically identical with the *Dreadnought* (Fig. 275) when using her cruising turbine, and if H.P. Main be







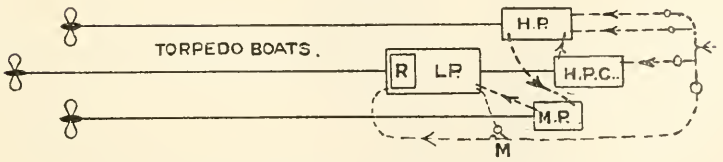


FIG. 277.—Three-shaft Arrangement of Direct-drive Turbines.

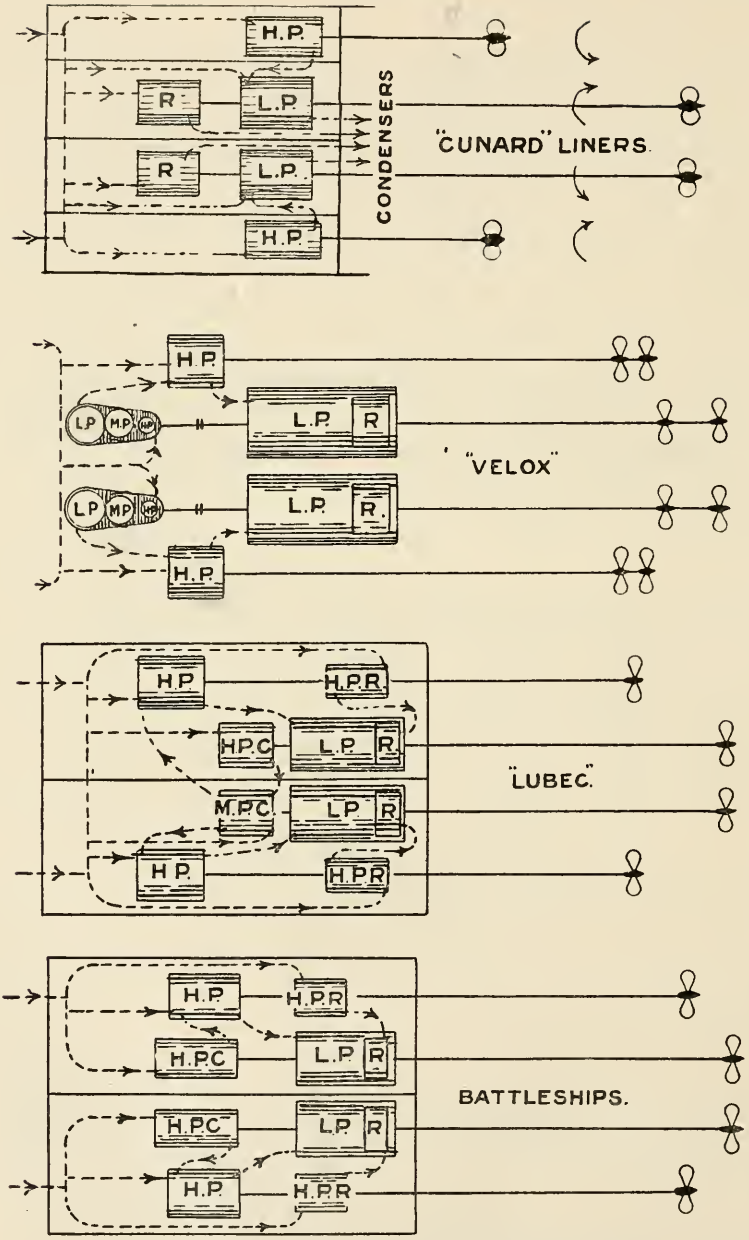


Fig. 278.—Four-shaft Arrangement of Direct-drive Turbines

substituted for H.P. Cruising, and M.P. Ahead be substituted for H.P. Main Ahead, merely a matter of names, the same figure shows either system. The difference is internal, and for cruising economy in the *Repulse* the overload system is relied on instead of the separate cruising turbine.

Single Reduction Gear.—Reduction gearing fitted in connection with the propulsion of vessels is almost always of the double helical type, the teeth in which are cut on an angle of about 30 degrees with the axis in opposite direction on the two sides as shown in the figures. The machining of these teeth requires meticulous care, and great accuracy is necessary to secure the high efficiency of about 98½ per cent; the other 1½ per cent is absorbed in oil friction, and any less efficiency than 98½ per cent denotes that the teeth are wearing and not properly cut or set. Thus if no wear takes place, and this is usually the case with well-cut gearing, it can be stated with a fair amount of certainty that the loss does not exceed 1½ per cent by friction in the gearing itself, with probably an over-all loss of about 2 per cent if the pinion and wheel bearings are included.

The number of teeth in the wheel should never be an exact multiple of the number in the pinion, otherwise any inaccuracy in, say, one tooth of the pinion, is likely to increase in dimensions owing to frequent contact in the same positions; by a trifling inexactitude any differences gradually grind away and a perfect contact becomes practicable at all points along the line. For example in the *Furious* the reduction is from 46 to 11 for the H.P., and from 86 to 11 for the L.P. turbine.

In **Mechanical Gearing,** either of the single or double reduction type, reliance is placed on a flexible coupling between the turbine and pinion and on the accuracy of machining the contact surfaces; the above remarks apply particularly to what is known as the *Parsons* system.

In the **Westinghouse Gearing** (Melville and MacAlpine patents) a hydraulic floating frame, for supporting the pinion bearings, and automatically regulating the alignment of the pinion to accord with the tooth pressure, is relied on to give great efficiency in combination with uniform high transmission pressures through the entire line of contact between the teeth in gear. It is claimed for this system that all objectionable noise is eliminated. This system is usually fitted in the United States in preference to the Parsons system.

In either system, the number of shafts is dependent on the power which can be transmitted in the form of pressure per lineal inch on

the pinion. The eventual success of either system also depends on the wear of the gear ; and it may be considered that success is assured with good machining and accurate adjustment. A good system of lubrication is essential to success.

The efficiency of the thrust mechanism is an important factor in the over-all efficiency of all systems of marine propulsion. In a direct turbine drive (without gearing) the effects of steam pressure differences and the direction of the steam travel may be controlled to balance, more or less, the propeller thrust, and only about 2 per cent of the S.H.P. may then be absorbed by the thrust mechanism.

In a geared system, both the steam thrust in the turbine and the propeller thrust on the main shafting must be balanced so as to bring no axial thrust on the gearing. The introduction of the Michell oil film system of lubrication and single thrust collar solves this problem in a simple and most efficient way, because the thrust loss is only $1\frac{1}{2}$ to 2 per cent instead of anything up to 15 or 20 per cent with the old multi-collar system, and a direct-drive whether for turbines (unbalanced), reciprocating steam or internal combustion engines. Absolute perfection is impossible of attainment; there must be some loss in all systems of transmission, and if the loss does not exceed $1\frac{1}{2}$ to 2 per cent, then practically the limit of high efficiency is reached as this loss is due to the friction of the oil film itself.

The maximum efficiency obtained by a direct turbine drive is seldom above 59 per cent for the propeller because of its high speed of revolution. By using gearing and reducing this rate of revolution it is practicable to increase the propeller efficiency to 69 per cent, and in a few instances a higher efficiency has been claimed.

Beginning with small cruising turbines connected by gearings with the main turbine shafts in two destroyers, and followed in other vessels, including a slow mercantile vessel, by direct gearing connection between the main turbines and the propeller shaft, experience proved sufficiently successful to " carry on," and during the war and since, the adoption of gearing has increased both in fast warships and in slow mercantile vessels.

Instead of a compromise in which the turbine is run somewhat below its most efficient blade velocity, and the propeller is run somewhat above its most efficient speed, gearing allows both mechanisms to run at the best speeds adapted to their efficiency ; the turbine runs at a higher blade velocity and the propeller runs at a slower rate of revolution. The effect is very marked and generally produces a

saving in steam and fuel of about 25 per cent over the direct drive. There is also some saving in weight and space, because the turbines are much smaller, about two-fifths, than in the direct drive, although the gearing is generally a very weighty article and the propeller shaft and propeller heavier for its slower rate of revolution.

Fig. 279 shows one of the four sets fitted in H.M.S. *Furious*, with its single reduction gearing, and the covers removed from the gear case and turbine casings or cylinders. Each set will develop upwards of 22,500 S.H.P. when the propeller shaft is making 330 revolutions per minute, and consists of one H.P. turbine (the rotor of which is shown with its shaft at the back of the picture) and one L.P. turbine at the front of the picture for working ahead and one for astern on the left in the same casing. The H.P. turbine makes 2580 revolutions per minute, and the L.P. turbine makes 1380 revolutions per minute when running at full power. In this arrangement, which is common to other vessels, each set of turbines is fitted in its own engine room, separated by water-tight bulkheads from the other compartments. But under normal working conditions the engines are worked as two units only for the four shafts as in the *Repulse*, and arrangements are made for the necessary steam and control connections in the two central engine rooms ; thus the working is actually the same as a twin screw vessel and simplifies matters on the bridge deck.

The relative size of the L.P. turbine rotors fitted in the *Repulse* with a direct drive and the L.P. turbine rotors fitted in a large high-powered cruiser with a geared drive, is shown in Fig. 280 ; the relative power being about 3 to 2.

In the large light cruiser *Raleigh*, built and engined by Messrs. Beardmore & Co., of Dalmuir, the turbines are of the Brown-Curtis type arranged for driving four propeller shafts through single reduction gearing. Each propeller is driven by a separate set of turbines of 17,500 S.H.P. consisting of one H.P. and one L.P. turbine on separate turbine shafts gearing into the propeller shaft wheel. There is also a cruising turbine fitted at the forward end of the H.P. turbine, and fitted with a claw-clutch for disconnecting when not required. The four sets of turbines are arranged in two engine rooms, the forward one containing the wing and the after one the central turbine sets, as shown in Chapter XXIV. ; for astern working, an astern turbine is fitted inside each L.P. casing. Michell thrust blocks are fitted to all shafts.

The trials of the two Swedish destroyers (Figs. 262, 263) compared

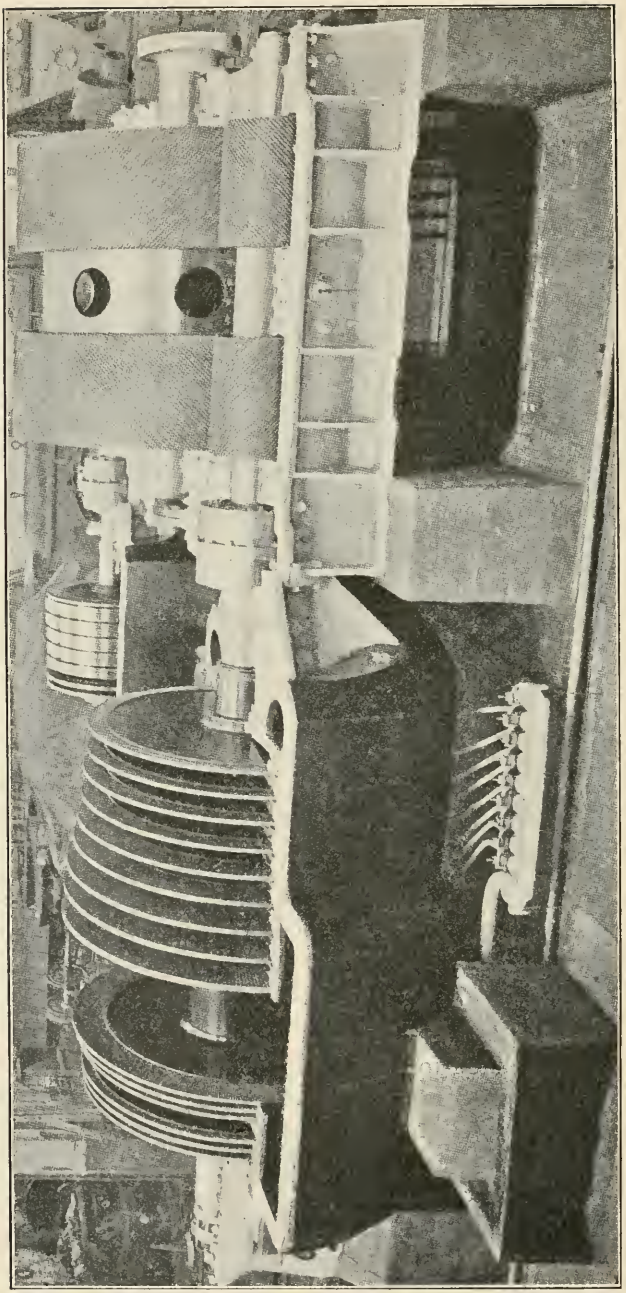


FIG. 279.—Single Reduction Gearing, H.M.S. *Furious*.

with the results obtained with others of the same class, but fitted with reciprocating engines or with direct-driven turbines, give an additional proof of the economy of geared turbines and the consequent increase of power and speed obtainable, although in this case the displacement shows an increase from 350 to 439 tons, of which about

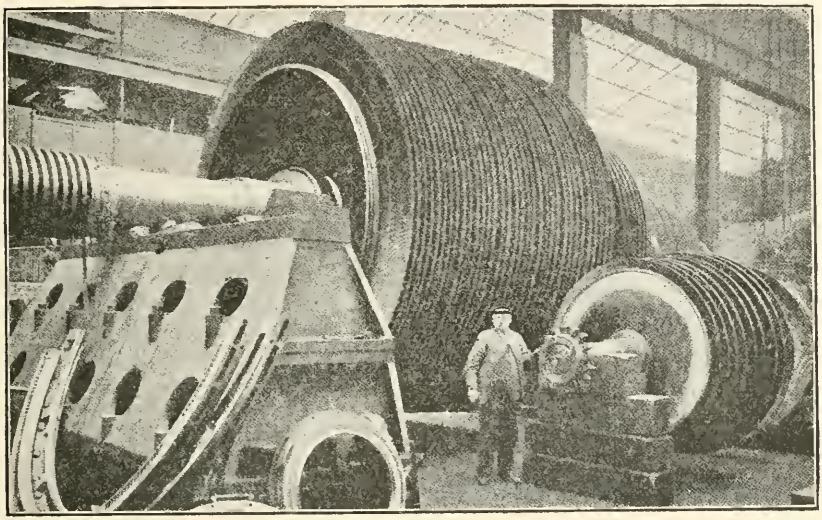


FIG. 280.—Comparison of Size of L.P. Turbines, Direct-drive and Geared. Power ratio 3 to 2.

15 tons is due to the machinery and its greater consequent power. The boilers are identical in each boat.

The coal used in the *Wrangel* trials was not so good, owing to war conditions, as in the other boats' trials, and in other details possibly the comparison is not exactly accurate, but for all practical purposes it may be accepted as a fair comparison. On the *Wrangel* cruising trials the steam consumption of the auxiliaries was unexpectedly high, 10,344 lb. instead of the 3307 lb. per hour expected; this was principally due to the rotary air pumps, which took practically the same amount of steam as at full power, but as the exhaust was used in the main turbine the latter benefited.

[TABLE

		"Vale" Reciprocating Engines.	"Hugin" Direct Driven Turbines.	"Wrangel" Single Reduction Geared Turbines.
Trials at Full Power				
Displacement	Tons	350	380	430
Shaft Horse-Power		7050	9500	11,000
Revolutions per minute		386	800	450
Speed	Knots	29·4	30	34
Steam per S.H.P.-hour	Lb.	19·40	13·23	11·46
Propeller efficiency	%	64	50	61
Steam per effective H.P.	Lb.	32·71	31·08	22·04
Coal per effective H.P.	Lb.	4·80	4·42	3·24
Total weight machinery per S.H.P.	Lb.	51·8	38·9	34·62
Total engine-room weights per S.H.P.	Lb.	35·29	34·72	25·39
Trials at Cruising Speeds				
Shaft Horse-Power		700	950	1,000
Per cent of full power	%	9·93	10	9·09
Net Steam per S.H.P.-hour	Lb.	23·61	26·01	16·97
Propeller efficiency	%	70	61	67
Coal per effective H.P.	Lb.	3·92	5·64	3·41

Double Reduction Gearing.—For slow mercantile vessels single reduction is insufficient to bring the propeller speed down to that consistent with good propeller efficiency, and therefore double reduction gear is now becoming a common system for these vessels.

Messrs. Vickers, Ltd., have two designs particularly for this purpose: in one with *reaction* turbines and the other with *impulse* turbines, arranged with the gearing either in a single or in separate casings and allowing a considerable variety of turbine arrangement on the several shafts.

The gear wheel must necessarily be on the same level as the propeller, not necessarily exactly horizontal, but preferably parallel to the keel line; but the turbines may gear into the turbine gear wheel at any convenient level either at the axis or above or below the axis line.

Typical arrangements for Reaction turbines are:

(1) H.P. and M.P. turbines each developing $\frac{1}{4}$ power and driving the same first reduction wheel; L.P. turbine developing $\frac{1}{2}$ power and driving another reduction wheel. Both first reduction wheels engage a single second reduction wheel. The H.P. Astern turbine is fitted inside the M.P. (Ahead) casing and a L.P. Astern turbine is fitted in the L.P. casing.

(2) Instead of the above three turbine shafts there are only two;

the M.P. (and enclosed H.P. Astern) turbine being fitted on the same shaft as the H.P. but forward of it.

(3) In this the M.P. turbine (and enclosed H.P. Astern) is omitted.

Typical arrangements for impulse turbines are similar to (2) and (3) above.

All the above arrangements can be adapted for either one gear case for all the gear wheels (Fig. 283), or for a separate casing for each of the three-gear wheels, in which case the two first reduction wheels are not situated between the two parts of the second helical reduction wheels but placed forward of them.

The choice of arrangement must depend on the space available, but where space permits there should be little hesitation in selecting a single gear case and the impulse type of turbine, for which two cylinders are quite sufficient to give the necessary economy of expansion of the steam to its practical limit.

A limited number of installations of double reduction geared turbines were made by Messrs. James Howden & Co. during the war for standard ships. These turbines were of the (Swiss) Zoelly type, an impulse turbine of good reputation with one H.P. and one L.P. turbine, having a combined S.H.P. of 2300 divided equally between the two turbines. The turbines run at 3200 revolutions per minute, reduced by double reduction gearing to 71 revolutions per minute on the propeller shaft. By by-passing high pressure steam into an intermediate stage of the H.P. turbine the power can be increased to 2900 S.H.P. with a corresponding increase of revolutions per minute to 3500. The gearing helix angle is 30 degrees. The weight of the gearing and its parts is 33 tons and of the turbines 22 tons. There are no velocity compounded stages in these turbines, and each stage consists of one guide or nozzle wheel and one rotor wheel.

Westinghouse-Rateau Marine Turbines.—In Chapter XX. the Blading of this turbine was described with the general principles of its elementary construction, and its application to marine propulsion is next described and illustrated.

Generally there is one H.P. and one L.P. turbine in each set, mounted side by side, both working through high speed pinions on to double-reduction gearing and thus driving a single propeller shaft. This general arrangement is shown in Fig. 283.

Figs 281 and 282 show a cross section through these two turbines, and in each casing there is also an astern turbine.

For steaming *ahead*, the steam is expanded in nozzles *A* (Fig. 281)

to a pressure not greater than 30 to 40 lb. gauge, and then passes through a velocity compounded stage of two rotor wheels *B* into the four single impulse stages *C*. By suitable connections the exhaust from the H.P. turbine is carried into the L.P. turbine (Fig. 282) and is further expanded through the nozzles *D*, and six other single impulse stages into the exhaust space *E* in connection with the condenser shown below the turbine.

For steaming *astern*, the steam is first expanded in nozzles *F*

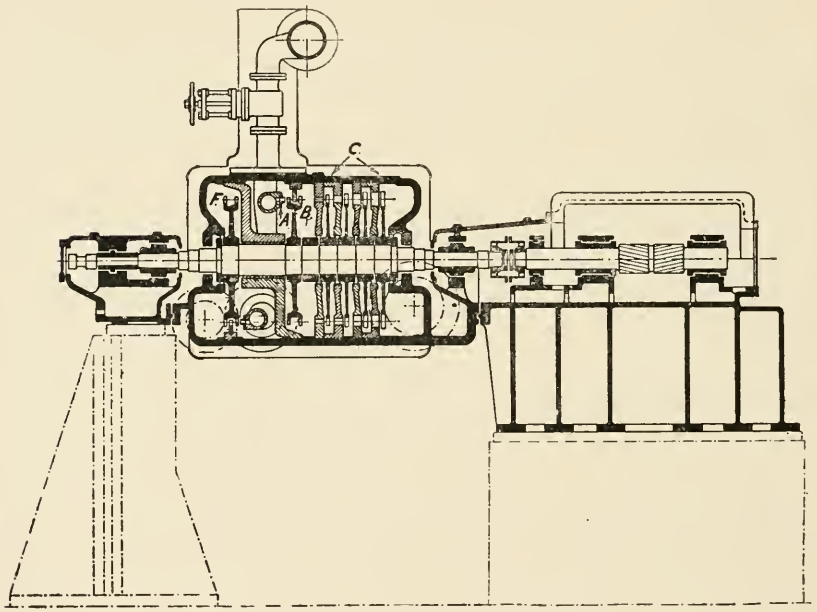


Fig. 281.—Westinghouse-Rateau H.P. Turbine.

(Fig. 230) and passes through a velocity compounded stage of two rotor rings of blading into the space *G* (Fig. 282) where it is further expanded through two single impulse stages into the exhaust space *E* and enters the condenser. About 70 per cent of the normal full power is available for astern working.

In the L.P. cylinder the pressure does not exceed a few pounds above atmosphere. Both cylinders are made of very high grade cast iron, and the *forward* ends are provided with sliding seatings to allow free expansion caused by variations of working conditions, and temperature changes.

The shafts are made of high quality steel, machined all over and stepped in diameter to take the rotor wheels, and as the shafts are short and stiff between bearings the critical speed at which vibration begins is well above the normal working speed.

Relief valves are provided for both the high and low pressure casings, to relieve any excessive steam pressure in case of emergency.

The *main bearings* are of the spherical type and made of cast iron lined with white metal. By means of liners of varying thicknesses, the main spindle can be set within one-thousandth of an inch to take

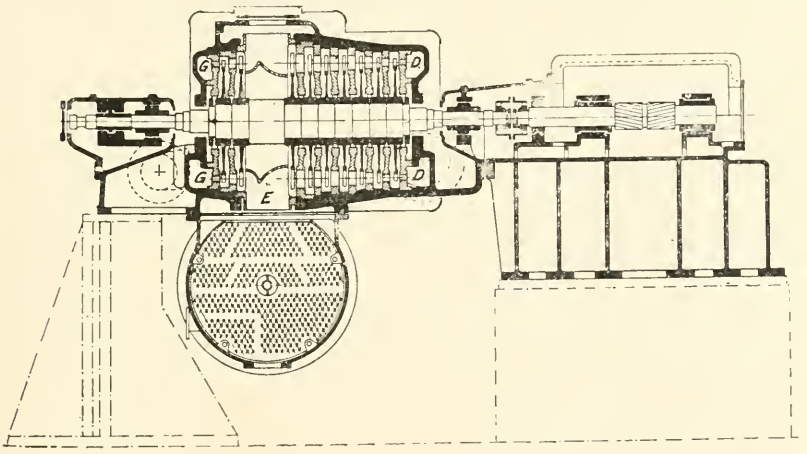


FIG. 282.—Westinghouse-Rateau L.P. and Astern Turbine.

up wear, etc. Special straddle gauges are provided for taking the adjustment.

The turbine *thrust block*, now generally of the Michell single-collar type, is used to maintain the correct position of the rotor spindle, but there is stated to be no axial or end thrust caused by the steam on the blading and moving parts because each rotating wheel is perforated with several holes to balance any difference of steam pressure on its two sides. Even if this statement is not absolutely accurate the steam thrust should be so small with a well-constructed turbine that it would be negligible.

A typical arrangement of double reduction gearing adapted to mercantile requirements is shown in Fig. 283 for a set of Westinghouse-Rateau turbines, described above, and is similar to that

fitted in the s.s. *Assiout* and *Amarna* (see *Engineering*, May 23, 1919). Each turbine runs at about 3000 revolutions per minute when developing the full power of 2500 S.H.P., and the propeller then makes about 70 revolutions per minute for 11 knots. The turbines therefore revolve at about 45 times the rate of revolution

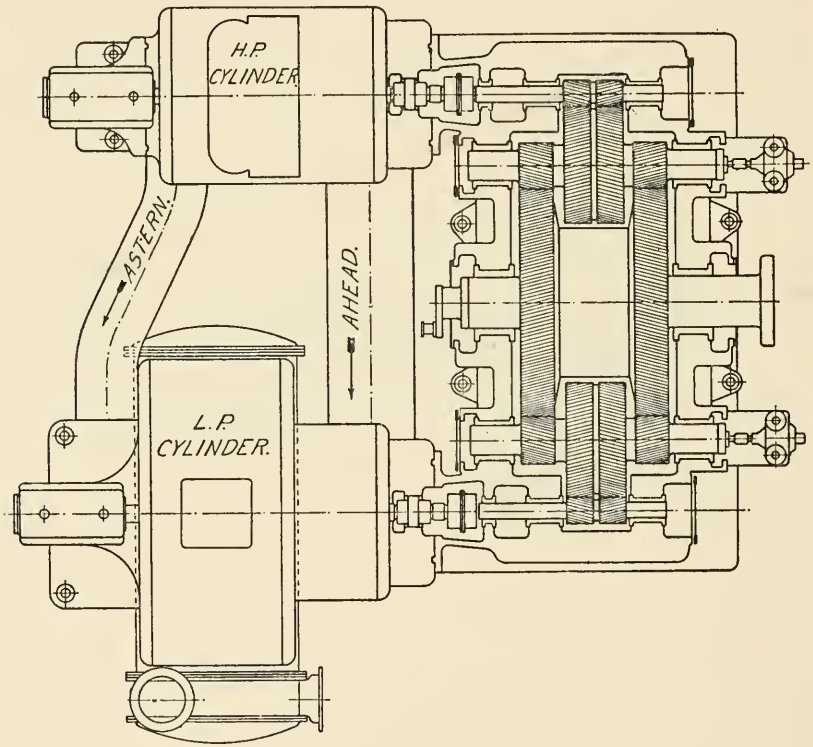


Fig. 283.—Double Reduction Gearing, Westinghouse-Rateau Turbines.

of the propeller. An astern section is fitted in both H.P. and L.P. turbines.

Fig. 284 shows the arrangement of turbines of a twin-screw merchant vessel, as designed by Messrs. Beardmore & Co. The gearing is of the double reduction type and fitted under one cover for each set of turbines. For ahead steaming there are one H.P. and M.P. turbine on one turbine shaft, and one L.P. turbine on another shaft. For astern steaming there are one H.P. Astern rotor inside the M.P. Ahead casing and one L.P. Astern rotor inside the L.P. Ahead casing. The main

condensers are fitted under the L.P. turbines, as shown in the longi-

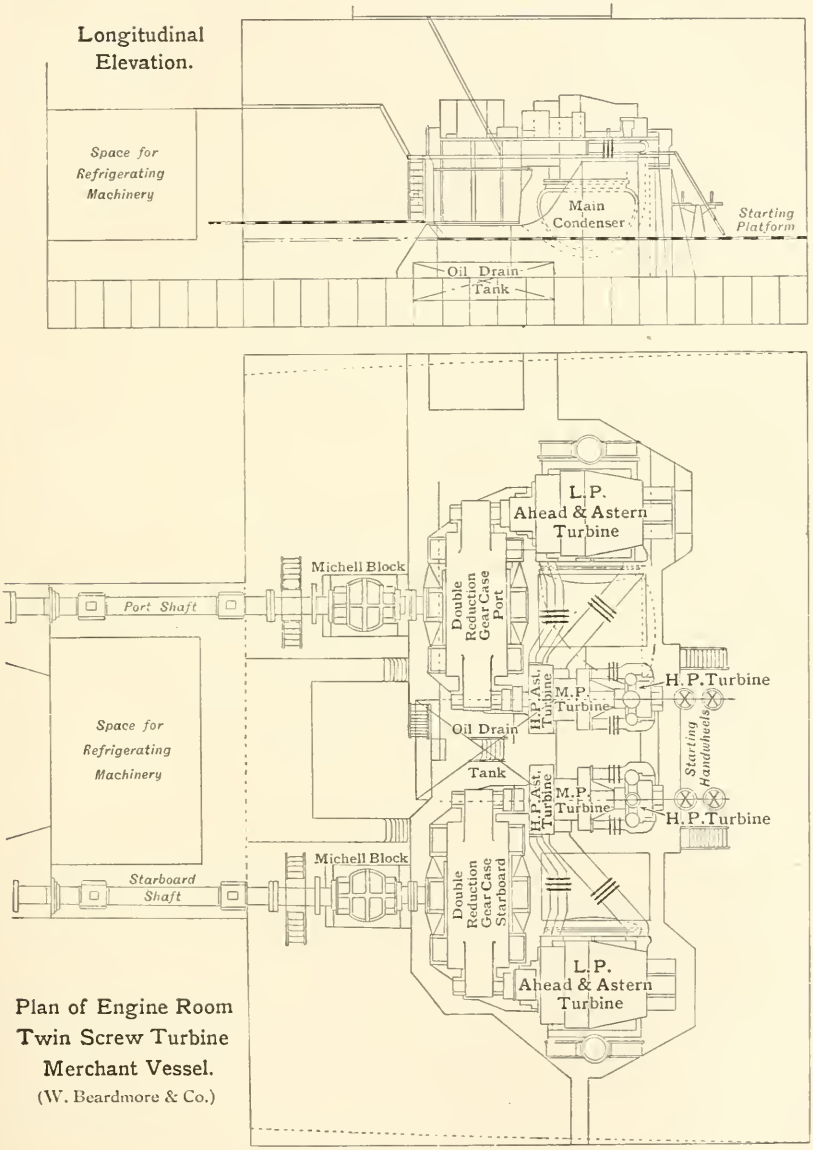


FIG. 284.—Arrangement of Twin-Screw Turbines.

tudinal view; this arrangement is becoming common and has been

frequently adopted for destroyers with single and double reduction gearing.

Further details of the general arrangement of auxiliaries, pipes, etc., of this installation are shown in Chapter XXIII. It would appear that the mercantile marine, which has been slow to adopt the turbine as a means of propulsion, has now been enabled by the development of mechanical gearing to obtain a good and economical system which should very shortly take the place of reciprocating engines in all new vessels.

In some gear drives for marine propulsion the Michell thrust is embodied in the same casing as the gearing, and the thrust block is of an improved pattern in which each pad tilts on a hardened steel ball, instead of on the end of a stud.

Brown-Boveri Geared Turbines for Cargo Steamers. — Among others, there are two standard sets for single-screw steamers between 2100 and 3000 S.H.P., and for 3100 to 4000 S.H.P. Twin-screw steamers may have two sets of either of the above. The standard propeller speed for both types is 75 revolutions per minute.

The design allows an overload of 10 per cent, and the S.H.P. astern is about 80 per cent of the normal full power ahead.

The weight of a single set of the smaller power is about 62 tons, and that of the larger size about 82 tons complete. The steam consumption is stated to be 9·5 lb. for the smaller power and 9·3 lb. for the larger power unit per S.H.P.-hour at full load with steam superheated to 470° F., at 200 lb. gauge pressure and 28 inch vacuum; or with dry saturated steam of the same pressure and exhaust at 28 inches the relative steam consumptions are 11 lb. and 10·8 lb.

In the arrangement shown in Fig. 285, the two turbines when going ahead revolve at 2980 revolutions per minute and the propeller at 72 revolutions per minute when developing 2400 S.H.P. Steam of 185 lbs. pressure is superheated to 320° C. or 608° F. and with a vacuum of 94 to 95 per cent a steam consumption of only 8·58 lb. per S.H.P. hour is claimed. The net weight of this set, including turbines, reduction gear, condensing plant, manœuvring valves and spare parts, is given as 75 tons.

The Ahead H.P. turbine is of the impulse reaction type, and the H.P. Astern turbine is a two-row impulse wheel. The Ahead and Astern L.P. turbines are of the reaction type.

Two oil pumps are fitted for each set, working in parallel, one driven off the pinion of the second gear and the other from the auxiliary

condenser pump, so that there is always oil in the bearings even during stops, etc., and if the auxiliary is stopped for any reason the oil is supplied from the direct-driven oil pump.

The 10 per cent overload is obtained by opening an additional set of nozzles in the H.P. cylinder and not by by-passing into the L.P. cylinder.

Fottinger Hydraulic Transmission and Reverse Gearing.—The Fottinger Transformer is capable of being used as reversing gearing in a direct-drive arrangement in which the transformer is, however, also a reduction gearing, and it has the further advantage

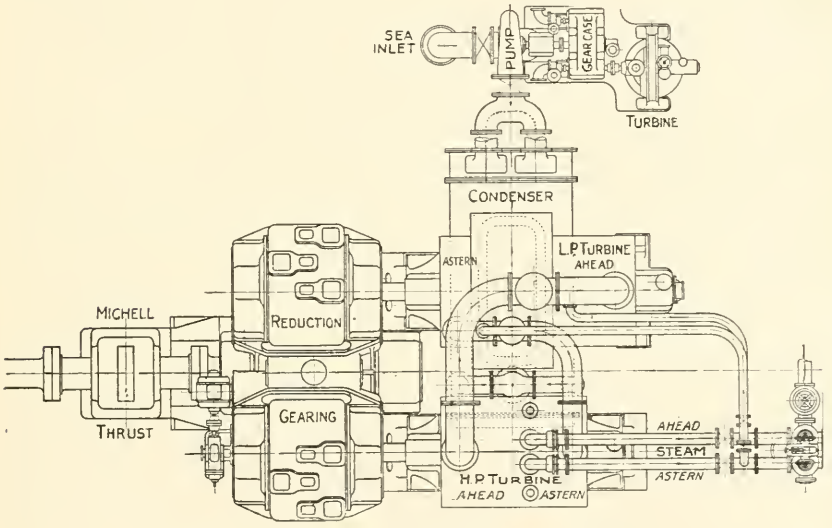


FIG. 285.—Arrangement of Turbines in Merchant Vessel.

of being capable of usage to balance the propeller thrust. The high limit of efficiency of transmission appears to be reached when the reduction ratio is about 6 to 1, but there appears to be no limit to the power which can be transmitted through a single shaft. The transmission loss may be still further reduced by utilising for feed heating purposes that proportion of the friction, from 3 to 4 per cent, which is converted into heat and absorbed by the water contained in the transformer. From experiment the best working temperature of this water appears to be about 170° F., which is a useful contribution for feed heating. The pressure at the pump discharge used for keeping the transformer charged with water is about 57 lb. per square

inch, and by suitable adjustment the temperature and pressure are regulated at about these figures when running at full power.

Compared with mechanical gearing the possible efficiency is as about 93 to 97 per cent for single reduction gearing, but mechanical gearing has the further advantage of being capable of speed reductions much greater than the 6 to 1 of the Fottinger hydraulic system. It may be concluded that only in special cases of vessels of 18 knots speed and upwards the Fottinger system has any real possibility of a favourable comparison in any of the essential qualities of economy of fuel, weight, and space.

As a German invention, its development during the war is practically unknown, but any increase in efficiency is not probable, and therefore the system as applied to the *Königin Luise* shown in Fig. 286 may serve for description although now some years old.

There are two distinct sections of the apparatus, one for ahead and one astern working, and it thus suffers from the same disability as the turbine. The whole interior of the transformer is charged with water by a pump in the first instance and a pump is used to keep it fully charged. An arrangement for cooling or changing the water is necessary to carry away the heat generated in the transformation and which, as already stated, can be utilised for feed heating.

The Ahead primary wheel *D* is keyed on to the turbine shaft which is coupled at *A*. The No. 2 secondary wheel *F* is keyed on to the propeller shaft with its thrust collars shown at *Q*. The Ahead primary wheel *D* cannot be seen because it is concentric with and revolves inside the No. 1 secondary wheel *E* which is bolted to *F* and therefore revolves with the propeller shaft. The water from the primary wheel *D* delivers up part of its energy to the vanes in *E*; and is then led through guide blades fixed in the casing into the secondary wheel *F*, where it gives up more of its energy and then returns to the suction side of the primary wheel *D*, where it is again energised by the action of the turbine through the medium of the wheel *D* and all the operations are repeated.

The astern portion of the transformer gives about 70 per cent of the ahead power and is smaller and differently arranged. The Astern primary wheel *B* is contained in a separate casing forward of the ahead casing and *B* is keyed on to the turbine shaft. This wheel *B* discharges through blades which are fixed to the casing and are set so as to reverse the direction of flow into the secondary Astern wheel *C* which is bolted to the first secondary Ahead wheel *E* and through

this is connected with the propeller shaft which it drives in the astern direction. At all times the turbine runs in one direction of rotation

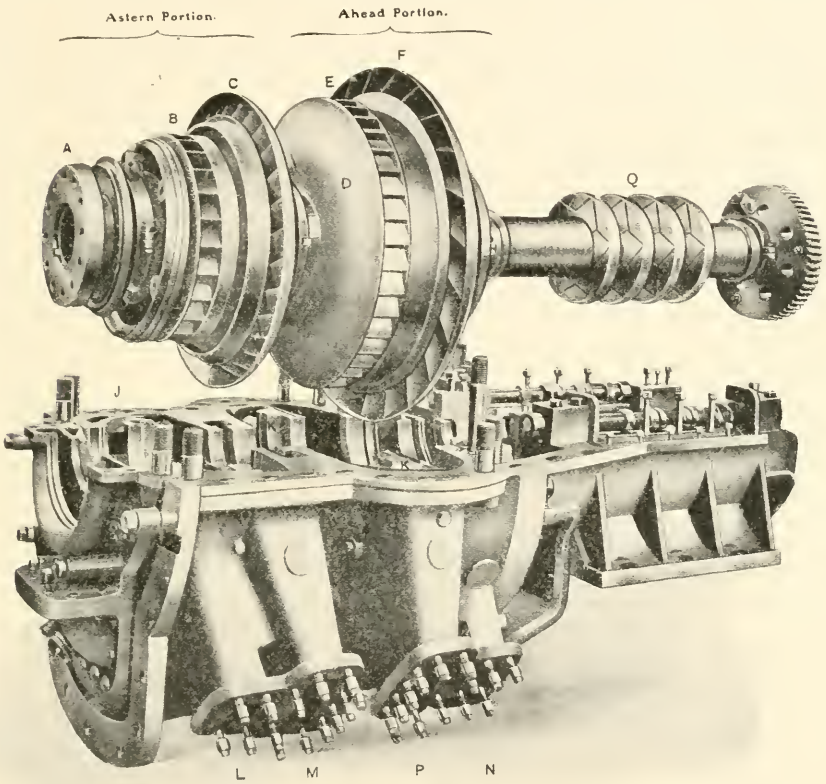


FIG. 286.—**Astern** and **Ahead** Fottinger Hydraulic Transformer.

A, Turbine Coupling.
B, Astern Primary Wheel.
C, Astern Secondary Wheel.
D, Position of Ahead Primary Wheel, which cannot be seen in illustration because it is concentric with, and revolves inside, No. 1 Ahead Secondary Wheel *E*.
E, No. 1 Ahead Secondary Wheel.
F, No. 2 Ahead Secondary Wheel.
J, Fixed Astern Guide Wheel.
K, Fixed Ahead Guide Wheel.
L, Entry of Water from Discharge Pump—Astern part.
M, Exit of Water—Astern part for manœuvring purposes.
N, Entry of Water from Discharge Pump—Ahead part.
P, Exit of Water—Ahead part for manœuvring purposes.
Q, Thrust Collars on Propeller Shaft.

only and it can be kept running at a constant speed, which is controlled by means of a governor.

The Make-up pump is of the centrifugal type, having its shaft

vertical, and is driven by a small independent impulse steam turbine. The pump chamber is submerged in the tank which supplies water to the transformer. The pump discharges through the main manœuvring valve into either the Ahead *N* or Astern portion *L* of the transformer. Boiler feed is used for the transformer water supply, and is being constantly heated by the work done upon it in the transformer. The ports in the manœuvring valve chest are so arranged that when the primary ahead pump chamber is open to receive water, the astern chamber is open to the drain tank, and *vice versa*. When the manœuvring valve is in its mid position, both ahead and astern chambers are emptied together and it is impossible to fill both at the same time. The Make-up pressure is regulated according to the number of revolutions of the propeller required ; when slowed down the water wheel cavitates, that is, the water contains a certain proportion of air and there is a corresponding slip.

Thrust blocks are fitted to take the turbine thrust and also to take the propeller thrust, but as the apparatus and the turbine are already fairly well balanced against such thrusts these blocks are more in the nature of adjusting arrangements.

Electrical Transmission.—Two systems of electrical transmission are practicable : (1) Entirely electrical, in which the generator and motor are not necessarily connected by shafting, or (2) partially electrical, in which the generator and the electric motor are on the same line of shafting, and the generator can be used for a direct drive on the propelling shaft.

In many submarines, probably the great majority, an internal combustion engine and an electric motor, also used for generating electric power, are fitted on the same propeller shaft. On the surface the internal combustion engine can be used either for propelling the vessel or for simply driving the electric generator for storing up electric energy in storage batteries. Below the surface the electric generator is used as a motor for propelling the submarine so long as the batteries are sufficiently charged for the purpose of supplying power. For starting the internal combustion engines, the electric motor is also used.

In the " K " class submarines of 24 knots surface speed, it was necessary to install steam turbines instead of internal combustion engines on account of the greater power required. The electric drive is fitted for use when under water, and in addition a Diesel engine is provided for use just before diving or immediately after breaking sur-

face, to shorten the time of diving or of getting under way quickly after coming to the surface. The transmission from the Diesel engine is through the electric motors, so that these vessels have not only

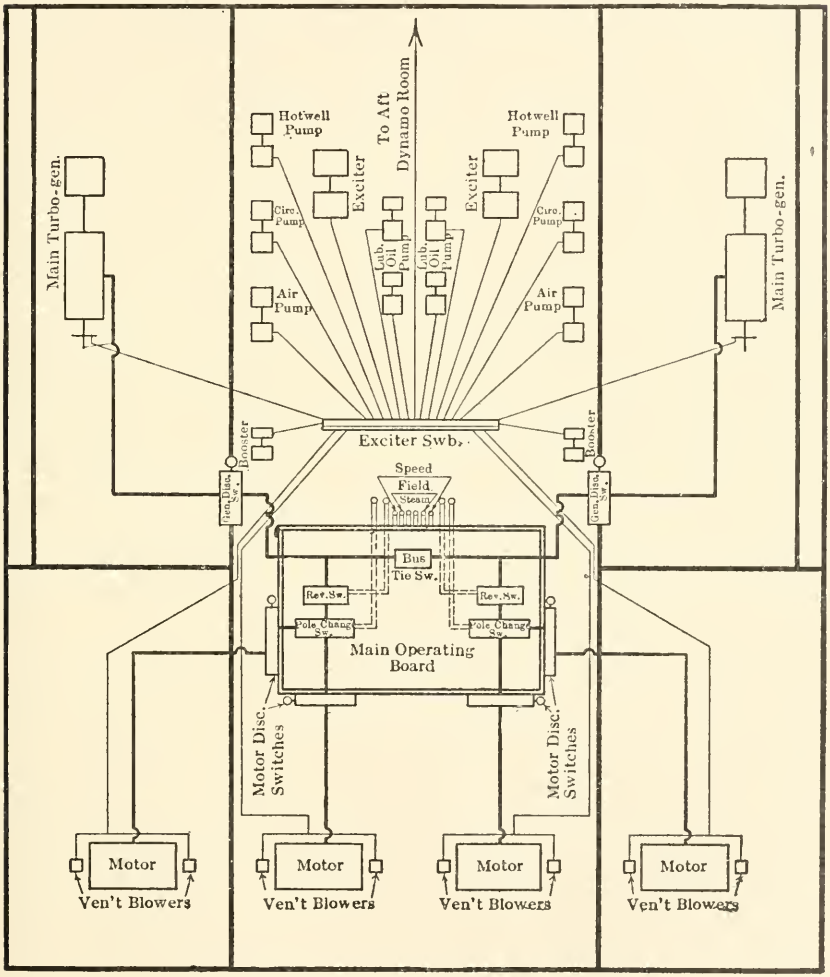


FIG. 287.—General Arrangement U.S.S. *New Mexico*.

geared turbines for the steam drive, but they have also electric transmissions for the Diesel engine and for the electric battery drive when under water.

Two merchant vessels, the *Jupiter* and *Wulsty Castle*, have been

fitted with complete electrical transmission systems, and the first capital ship in any navy to be fitted with the electric drive is the U.S.S. *New Mexico*, with a designed speed of 21 knots and a shaft H.P. of 29,000.

In the *New Mexico* the power is derived from two turbo-electric generators (Fig. 287), and transmitted through four electric motors to four propellers. For 21 knots these electric motors make about 167 revolutions per minute and are reversible. The steam consumption at full power is about 11 lb. per S.H.P., increasing to 14 lbs. at about 3000 S.H.P., which shows no advantage over the single reduction by mechanical gearing as used in the British Navy. The *Naval Annual*, 1919, puts the weight of the *New Mexico* installation at 2350 tons or about 170 lb. per S.H.P., whereas the machinery of corresponding power of British warships works out at about 140 lb. per S.H.P.

In the *Tennessee* and other U.S. battleships there are two turbo-generators of 12,500 Kw. each, capable of speed variable between 1500 and 2250 revolutions per minute. These deliver their power to three electric motors, each with a maximum of 8375 B.H.P. when running at 186 revolutions per minute. The motor stator winding is arranged so that the number of poles can be diminished from 36 to 24, giving a corresponding motor speed of 124 and 186 revolutions per minute when the turbines are running at full power and 2250 revolutions per minute. When the turbines are reduced to 1500 revolutions per minute the corresponding motor speeds are 84 and 126. By this arrangement the ship's speed can be varied from 21 knots to 9 knots. Below 15 knots only one generator is required. Intermediate speeds are obtained by variation of turbo-generator speed. In the later battleships it is proposed to increase the horse-power to 60,000.

The **Ljungstrom Steam Turbine** is not directly applied to marine propulsion but only indirectly through electric transmission to electric motors, which in their turn may drive the propeller shaft either directly or through single reduction gearing as in the s.s. *Wulsty Castle*, or double reduction gearing.

Fig. 288 shows one of a pair of these turbines as fitted in the s.s. *Wulsty Castle*. Each turbine drives two electric generators, which run in opposite directions of rotation corresponding with the Ljungstrom system, thus making four generators in total. Each generator is designed to develop 625 Kw., or 837·5 B.H.P., when running at 3600 revolutions per minute. Steam superheated to 625° F. and at 220 lb. gauge pressure is supplied from the boilers.

In another vessel the ratio of mechanical gearing reduction is 9 to 1, from 700 to 77 revolutions per minute, and speed regulation is obtained through electric liquid resistances. It is claimed for this system that the coal consumption is about one lb. per S.H.P.-hour, which is probably due to the turbine speed of about 3600 revolutions per minute.

Efficiency of Transmission Systems.—In all marine transmissions of power the various efficiencies are affected as well as the important factors of weight and space. Greater speed of revolution of any part of the plant means less weight, and this is particularly the

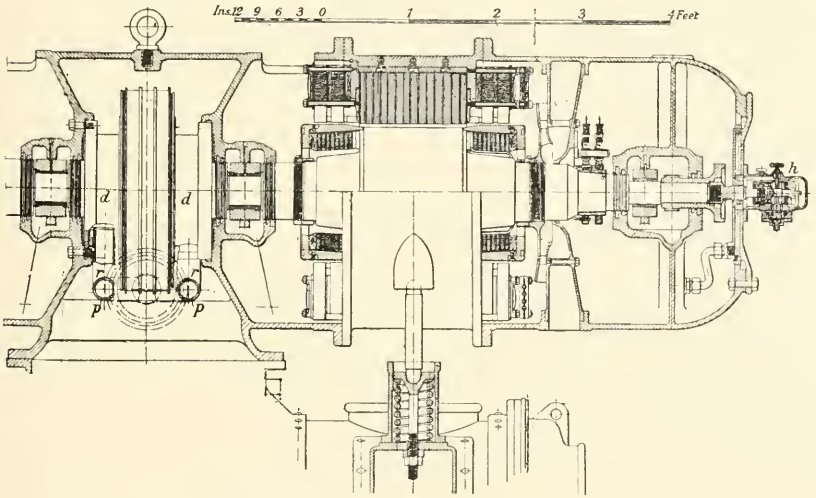


FIG. 288.—Ljungstrom Turbo-Electric Generators.

case with turbine machinery, in which by using gearing the revolutions have been very considerably increased, until in many instances a further increase would not be practicable with present materials and construction.

The propeller has the least practicable efficiency of all the elements constituting the total or over-all efficiency. To take a particular case, that of destroyers, in which the propeller efficiency with a direct drive seldom exceeded 40 per cent; by fitting single reduction gearing the rate of revolution of the propeller has been decreased from 20 per knot to about 10 or 11, with a consequent increase in propeller efficiency from 40 to 50 or 52 per cent. About 10 revolutions per knot appears to be the usual number for all classes of warships with reduction

gearing. In merchant vessels the number appears to be about 7 for the ordinary cargo steamer, with a slight increase on the lines of warships for the fast steamers carrying passengers.

The best speed at which to drive a propeller is dependent on the load draught and diameter which can be usefully employed, and therefore in some instances it is necessary to employ more shafts to obtain a suitable pitch and diameter corresponding to a high efficiency of propeller. The principal gain due to gearing in the over-all efficiency of the propelling plant is thus due to the propeller and the choice of a suitable propeller or propellers.

In warships, by the introduction of gearing it is now possible to run the turbines at the highest practicable speeds contingent on the type of blading and the centrifugal forces which limit the velocity of the moving blades. Better and more efficient forms of blading have recently been introduced as stated in previous chapters, and any advance in the present rates of revolution of turbines will be due to such improvements rather than to any other cause. For this reason the extended application of double reduction gearing is improbable, because the rate of revolution of the turbine is now of the order of over 3000 per minute and any further increase is likely to produce blade troubles and possibly stripping. In other words, the maximum speed of turbine and minimum speed of propeller can now be obtained with single reduction gearing in all except the smallest warships.

In merchant vessels of slow speed double reduction gearing is almost a necessity to the required efficiencies, because a reduction of the rate of revolution of the turbine is necessary to about one-fortieth or one-fiftieth at the propeller shaft.

The Fottinger hydraulic transmission does not permit these great reductions, and from the nature of its construction and its medium of transmission, *water*, it cannot give such good results. Even at its best ratio of reduction, 6 or possibly 7 to 1, its efficiency is less than mechanical gearing either of the single or double variety.

Electrical transmission saves no weight and space in large warships, and in the *Wulsty Castle* single reduction gearing is used between the electric motor and the propeller shaft; it is thus at best a hybrid system. A slowly revolving electric motor has all the disadvantages of a slowly revolving steam engine or turbine and involves great weight and space. Its paper efficiency may be equal to, but in practice it has proved less efficient than, mechanical gearing. Electrical gearing

also introduces a new element in personnel which may lead to difficulties and a probable increase in number and the wage bill.

For all transmission systems an improvement in either of the elementary efficiencies of the propeller or of the prime mover, whether steam or internal combustion engine, affects favourably the boiler plant or fuel consumption, which can thus be proportionately reduced in weight and space, as can also the accessories in a steam plant such as the condenser, pumps, etc., used in conjunction with the smaller quantities of fuel and steam.

Some interesting figures are given by M. R. J. Walker in a paper read before the British Association, 1919, on the Development of Geared Turbines, and various particulars relative to their efficiencies. A table of comparison of results, obtained in practice over extended periods for steamers of 4350, 10,000, 13,500, and 15,600 tons, show fuel economies from 15 to 19 per cent approximately, and in the two larger steamers a slight increase of speed in addition. The comparison is made between reciprocating engines and single reduction geared turbines; with double reduction gear Mr. Walker estimates a further economy of fuel of about 7 per cent.

Electric transmission has been adopted for many trawlers in connection with two 250 B.H.P. Diesel engines, which drive two electric generators which are connected electrically with a single 500 B.H.P. electric motor, driving the propeller at 200 revolutions per minute at full power. The advantages claimed for this system is the long period for which the vessel can be propelled at slow speed, and it provides for working the trawl winch.

PART VII

THE STEAM, CONDENSER, AND FEED WATER SYSTEMS

CHAPTER XXIII

THE STEAM, EXHAUST, AND DRAIN SYSTEMS

In earlier chapters the systems of generating steam in boilers were described with the necessary mountings for safe and efficient working, and the general arrangements of engines, both reciprocators and turbines, were also described with their internal fittings. In this chapter it is proposed to describe and illustrate the pipe and valve connections between the boilers and engines, with some of the special fittings necessary to them.

Steam and Exhaust Pipe Systems.—Generally there are two distinct systems of steam pipes, the main and auxiliary. These have been further complicated by the introduction of superheating, and for certain auxiliaries which are reciprocating engines it has been found desirable to retain the use of non-superheated steam, and at some point before the superheated steam reaches these engines the steam is mixed with non-superheated steam in sufficient proportion to prevent damage to the internal working parts of the engine. This damage could be prevented in another way, that is, by supplying internal lubrication oil or grease, but this would produce inefficient working of the boilers from deposits of grease and other matter attached to it on the boiler heating surfaces. With turbines no internal lubrication is necessary, and therefore superheated steam is preferably supplied without admixture of boiler steam and results in considerable economy. Incidentally this points to the abandonment of reciprocating auxiliaries

in favour of small turbines, and it will be noticed that a considerable advance has been made in this direction.

On each main boiler there is a main steam valve of the self-closing type preferably, and generally. These valves admit steam to the main steam system, which is generally arranged with one main fore and aft pipe in the boiler rooms on each side of a twin-screw vessel; if there are two boiler rooms the, say, port main pipe collects the steam by cross connecting pipes from the after boilers, and the starboard pipe then collects the steam from the forward boilers. Each main pipe enters the engine room through the bulkhead and a self-closing bulkhead valve is fitted on the end of each pipe in the engine room. The two bulkhead valves are connected by a cross connecting pipe on which the supply valves for the engine or engines are fitted. From this cross connecting pipe one or more connections with (self-closing) valves are made with the auxiliary system, if one is fitted, which is usually the case in high-powered vessels.

In some cases all, and in others only a proportion, of the boilers are fitted with auxiliary steam valves, but the systematic arrangement is similar to the main run of pipes and on a higher level if practicable.

The exhaust system from the auxiliaries follows the same lead as the steam supply, and if practicable the exhaust is on a lower level than the steam supply so as to allow a certain amount of drainage. Also both the steam and the exhaust systems slope downwards towards the engine room to give a natural flow of drainage to some predesigned position whence it can be taken away into the drain system or the condenser.

The systems are illustrated by a series of figures for a merchant vessel fitted with twin-screw turbines in Figs. 289, 290, and 291, and for a cruiser fitted with four-screw turbines in Figs. 292 and 294.

Fig. 289 shows an arrangement of three single-ended and three double-ended boilers, all fitted with superheaters. The main steam from the forward boilers and the middle after boiler is collected into one pipe, and that from the other two boilers into another pipe. This figure also illustrates the connections for the superheaters and boilers. In connection with each main steam valve there is one valve and pipe connection with the superheater and another with saturated steam, or direct boiler steam: so that either entirely superheated, or entirely saturated, or mixed steam can be supplied from each boiler. The middle forward single-ended boiler and the two wing double-ended

boilers are also fitted for supplying superheated steam to the auxiliaries, many of which are of the turbine variety in this vessel.

The considerable bends in the various pipes, as illustrated, are made for the purpose of allowing for independent expansion, which is very great.

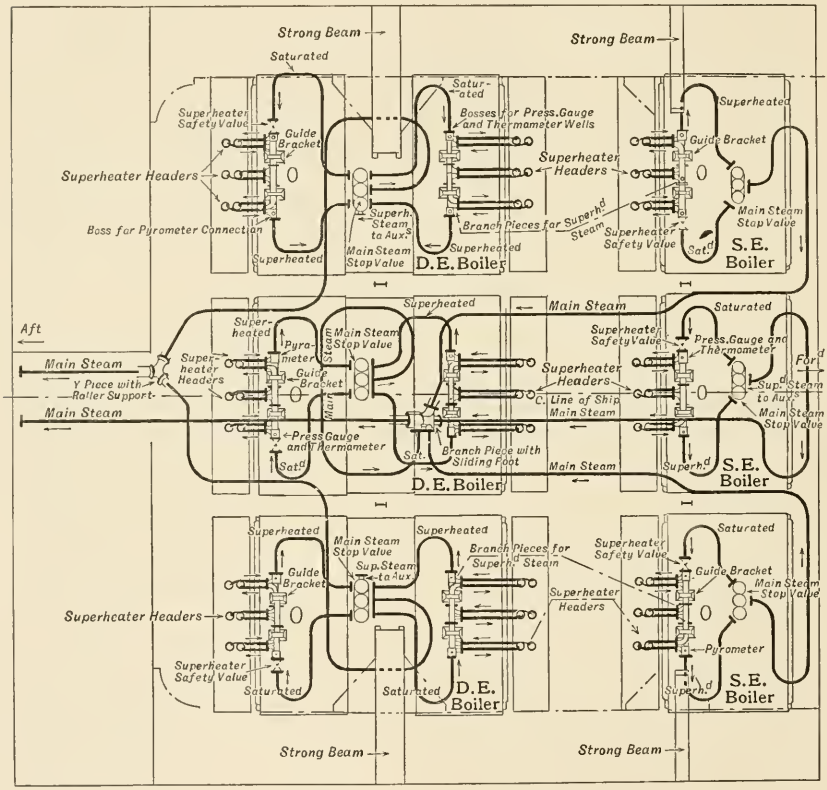


FIG. 289.—Steam and Superheated Steam Pipes in Mercantile Boiler Room. (Beardmore & Co.)

Fig. 290 shows the auxiliary supply system for the same vessel as Fig. 289. There are three supplies to the engine room : *Independent steam* which is saturated, or direct boiler steam from the port forward boiler ; *saturated steam* from the forward midship and three after boilers ; and *superheated steam* from the forward midship and two after wing boilers. These three supplies are known as the Independent, the Auxiliary, and the Superheated systems.

Fig. 291 shows the arrangement continued into the engine room, where it should be noticed that a connection is made with the main steam system. For clearness the main turbines and other parts are only shown in outline. With the arrangement as here shown it should be practicable to trace the various lines of supply, distinguishing those

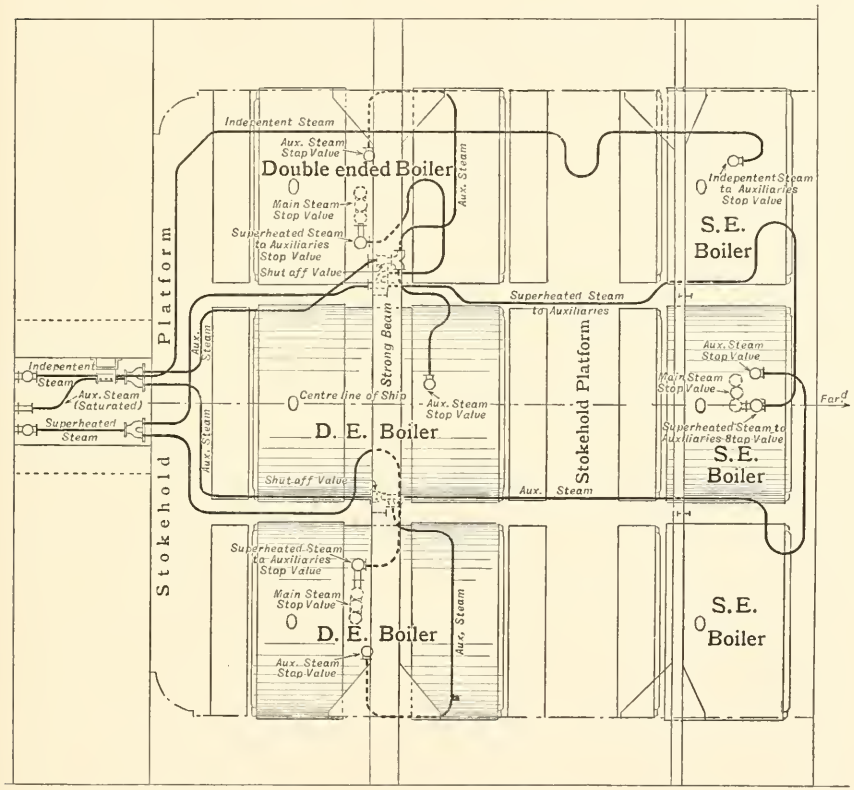


FIG. 290.—Auxiliary Steam System in Boiler Room of Merchant Vessel. (Beardmore & Co.)

which can utilise superheated steam without any harmful effect and the others to which a supply of superheated steam would upset their internal organs.

For the purpose of illustration of the arrangement in a cruiser of recent date the forward boiler room only is shown in Fig. 292. There are four boilers, and each is fitted with both a main, not shown, and an auxiliary steam valve. A branch pipe leads to the forward end of

the ship and is followed by the consequent exhaust run. These boiler rooms are a network of pipes and valves, and with their numerous auxiliary engines they are a complicated proposition, as will be clearly understood on studying the figures. But although this complication is apparent on paper, especially in a small illustration in a book of

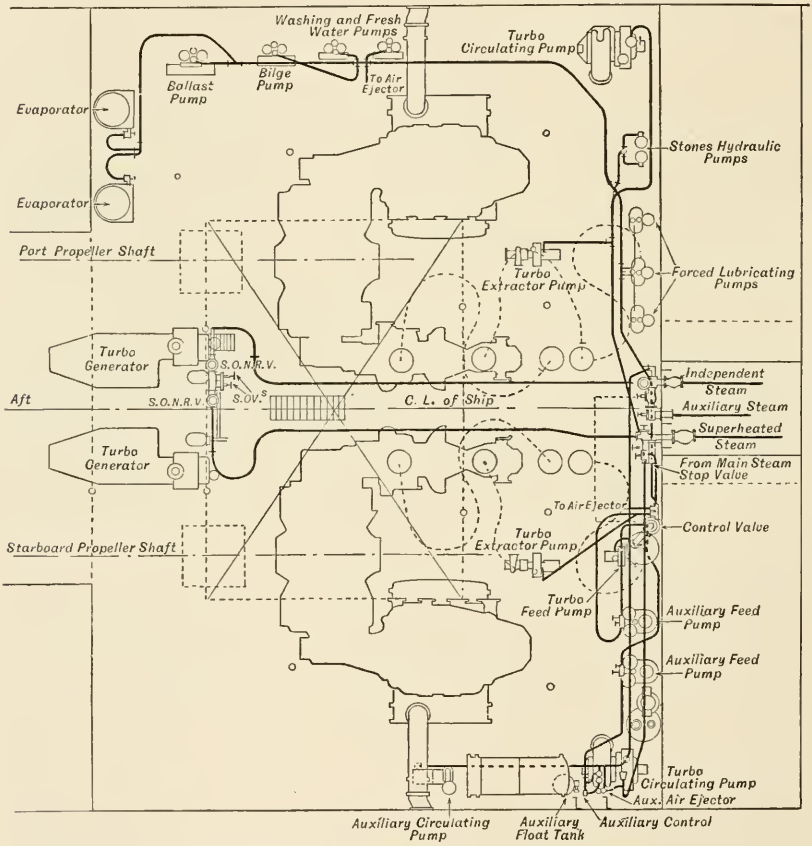
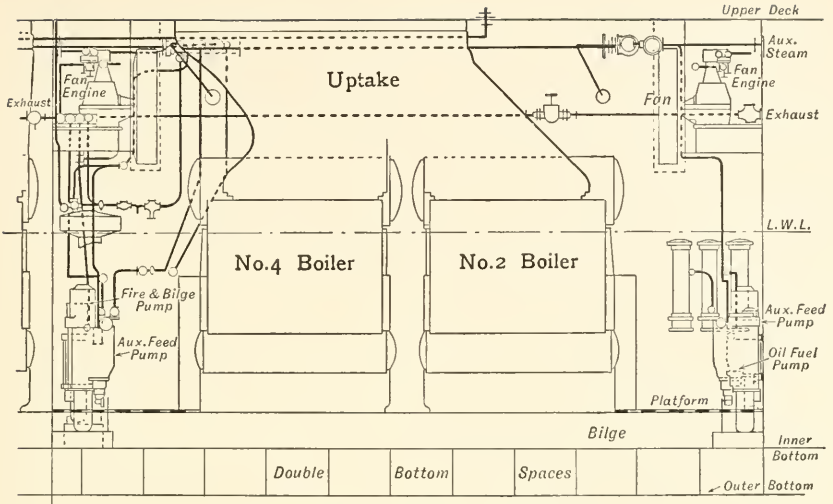


FIG. 291.—Auxiliary Steam Pipes in Mercantile Engine Room. (Beardmore & Co.)

this nature, there is a very distinct system running through the whole arrangement, and it is not so difficult to unravel after a careful study of the lead for any particular engine. One matter which is of importance in this connection is that the main feed pumps have both an auxiliary and a main steam supply, the auxiliary feed pumps have only an auxiliary supply. These figures, however, are not given for



Forward Boiler Room - looking to Port

Forward Boiler Room - Plan below Upper Deck.

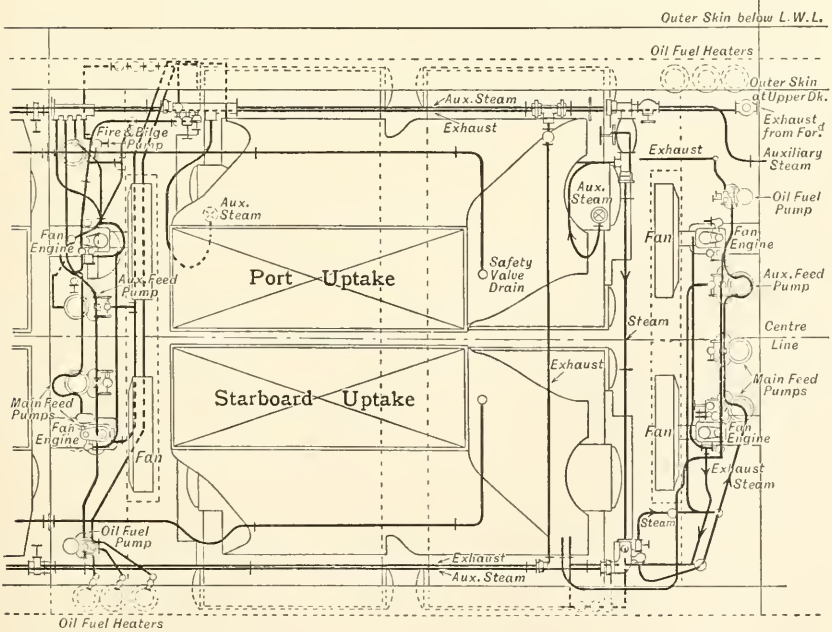


FIG. 292.—Auxiliary Steam System. (Beardmore & Co.)

any other purpose than a general idea of the general run of piping and its arrangement in a general way of what may be looked for in a warship of recent date. In the boiler room itself the pipes are nearly all in sight from the stokehold platform, and their run is fairly easy to grasp in detail. Every one who knows his ship must be able to locate any particular pipe or valve, and the study of the system is one of an engineer's first duties as soon as he arrives on board.

In Fig. 294 the engine rooms of the same vessel as illustrated in Fig. 292 are shown, with the positions of the main turbines and auxiliaries and the run of the main steam pipes and valve connections. The auxiliary steam pipes have been left out for clearness, but still the system seems unduly complicated. The arrangement is, however, systematic and quite simple when seen from the engine-room platform, and one soon gets to know the run of every pipe and its use.

An entirely separate system of valves and piping for the auxiliary steam service is preferable, because in case of accident to the main steam pipe, the auxiliary can then be brought into use ; and there are other practical advantages. About twenty years ago one of the main boiler steam pipes burst in H.M.S. *Porpoise* when entering a dangerous passage of the Tonga Islands ; by working through the auxiliary steam service in connection with the main service in the engine room the ship was navigated successfully to the anchorage.

From the auxiliary steam and exhaust service branches are led to each separate auxiliary engine as shown in the figures. These auxiliaries for the cruiser are enumerated and classified in Chapter XXIX., where they are also described.

In the engine room the auxiliary exhaust pipe runs aft to the condenser, and a cross pipe with a bulkhead valve connects the condenser with the other engine room. When no auxiliary condenser is fitted, the cross connecting pipe joins up the two main eduction pipes through the bulkhead valve. When the main engines are not under steam, one of the main condensers, a circulating pump (connected with the auxiliary steam service), and an independent air pump are used. The last is not always necessary, especially in small vessels, as the condenser can sometimes be drained through a non-return valve into the feed tank ; but the use of the air pump is preferable, as the water can then be pumped into the grease extractor before its return to the boiler.

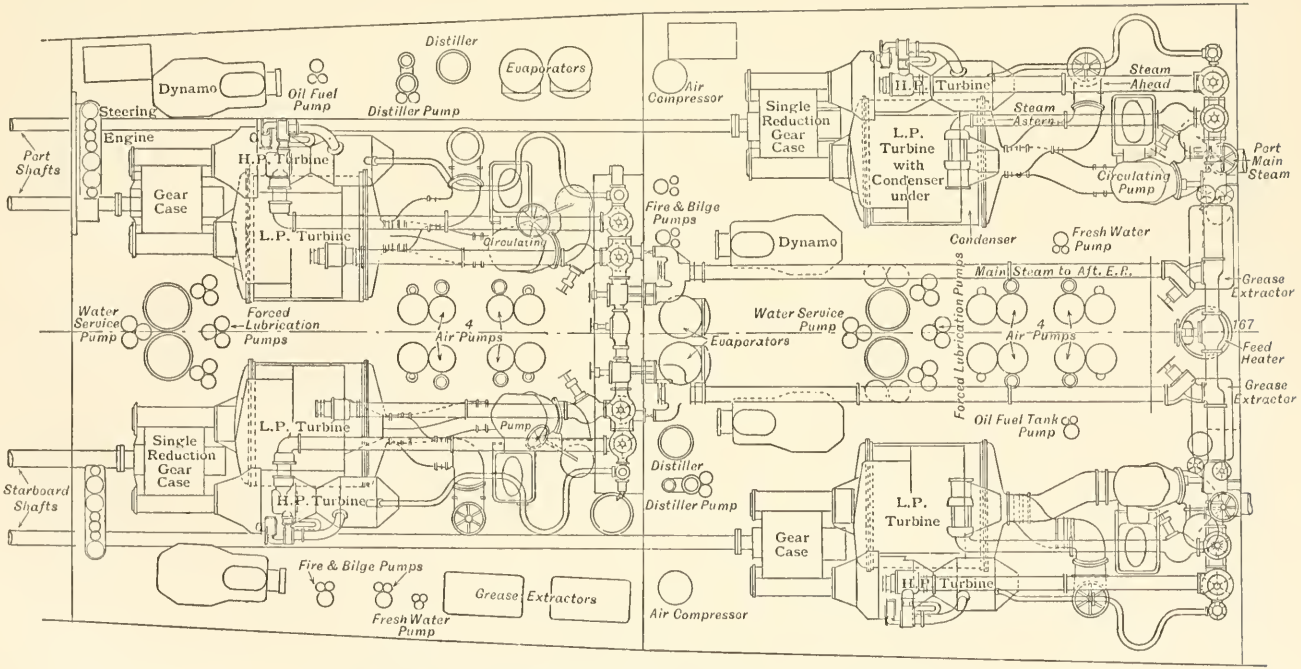


Fig. 294.—Main Steam System for Four Sets of Turbines. (Beardmore & Co.)

Closed Exhaust System.—Since 1897 many warships have been fitted with an economising device known as the closed exhaust system, and the arrangement of auxiliary exhaust pipes in Fig. 295 can be used in this way. The auxiliary exhaust is throttled at its connection, *T*, with the condenser, and a pressure of about 27 lb. per square inch is maintained in the exhaust pipes by means of a spring-loaded relief valve, *R*. Steam from this source can be used—

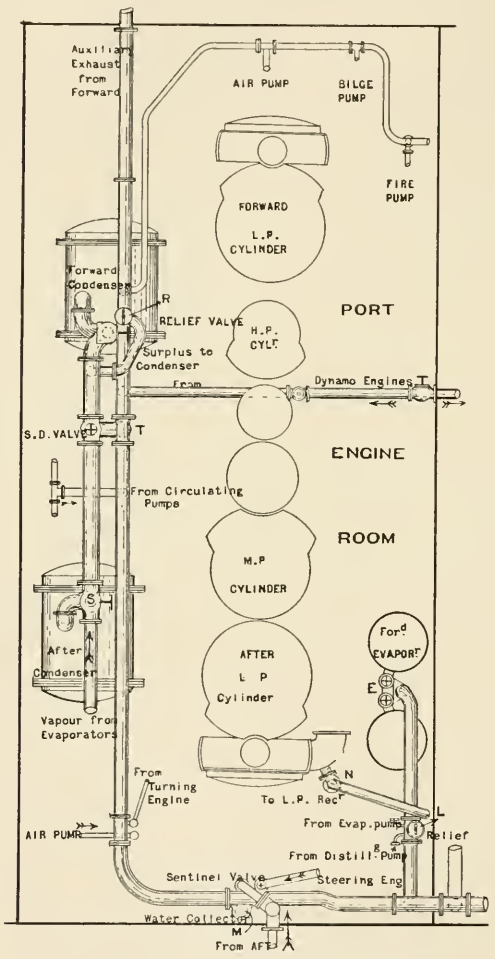


Fig. 295.—Closed (auxiliary) Exhaust System.

(*a*) In the L.P. cylinders or turbines, by passing through a relief valve in connection with the L.P. receiver, set at 25 lb. pressure, and a shut-off valve, *N*, workable from the starting platform.

(*b*) In the evaporators, by passing through a water collector, *M*, shut-off valves, *E*, and a spring-loaded valve, *L*, set at about 27 lb. pressure, into the evaporator heating coils. After doing work in the evaporator, in which the steam is condensed by boiling the water surrounding the coils, the condensed steam is passed into the condenser through the valve, *S*.

By using the closed exhaust system a large quantity of heat, otherwise wasted, is converted into useful work, and there is a considerable gain in economy. The back pressure on the reciprocating auxiliary

engines in connection with the system is considerably augmented, however ; but in some cases, where the system is used to supply small turbine auxiliaries there is a great economy of steam. In the more recently designed ships most satisfactory results have been obtained, and the practical gain is definitely marked.

The practical objection to the use of the closed exhaust is the water loss caused by increased leakage from the numerous piston, slide, and valve-rod glands throughout the auxiliary system. When a small vacuum can be maintained in the exhaust pipe, nearly all this leakage is prevented ; the feed pumps and other engines are better drained and more efficient ; and the evaporators are less used, with a consequent saving in steam and fuel. Some weight and complication is added by fitting the closed exhaust ; but it has justified its existence, and very economical results have been obtained—amounting to a production of 10 to 12 tons of fresh water in the evaporator for each 1 ton of coal burnt.

Expansion Joints on Main Steam Pipes.—The pipes conducting the steam from the boilers to the engines are long, sometimes about 100 feet, and the variation in length, due to their expansion when hot, is considerable. For ordinary temperatures and high pressures about $2\frac{1}{4}$ inches must be allowed for expansion in every 100 feet of length of steam pipe. The expansion may be allowed for in two ways—by making the pipes with bends of sufficient curvature, or by fitting expansion glands. The former method is generally used for pipes of small diameter or with sufficient space, and the latter for pipes of large diameter or where the space is restricted.

Each alteration of direction of the path of fluid pressure through a pipe produces a consequent end thrust or recoil which must be considered in connection with the allowance for expansion. For example, consider a pipe, with a right-angled bend, containing steam of 250 lb. pressure per square inch, and about 15 inches in diameter ; then the total pressure tending to force the pipe away from its end connections is equal to the pressure × area of the pipe = about 20 tons, for the figures stated above.

The necessary expansion where the space is restricted is allowed through expansion glands, and the end pressure is safeguarded by stays fitted between the flanges of the parts of the pipe. In Fig. 296 the end of one part of the pipe is made with a stuffing-box, *A*, into which the opposite end, *B*, of the pipe projects, and is made steam-tight by the packing gland, *C*, generally packed with alternate layers

of asbestos and elastic core packing. The unbalanced end pressure is taken on the guard stays, which are fitted through lugs behind the flanges or made in one with them. The check nuts hold one end of each stay fast, while the other can move through the holes within the limit of the bolt head. The check nuts are pinned through the stay, to prevent their moving or maladjustment after removal.

The ring *T* is a distance piece, and when necessary to test the pipes by water pressure, it can be removed and a blank flange substituted. Arrangements are made in a similar manner for the various pipes between the boilers and the bulkhead valves, the next length to the

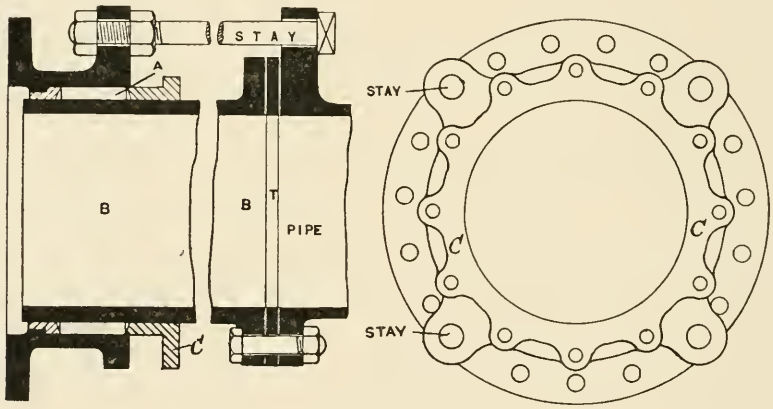


Fig. 296.—Pipe Expansion Gland for Straight Leads.

regulating valve, the various receiver and eduction pipes, and parts of the auxiliary steam service.

Ball Joints for High Pressure Steam Pipes.—A form of ball joint, originally known as Harter's joints, is shown in Fig. 297, and is one commonly used in high pressure systems where the lead is not quite straight. It is a modification of the one above described, and which is still used for low pressure steam and water.

Allowance must be made for longitudinal expansion and contraction under steaming conditions, and for this purpose the connecting ball is floated. In Messrs. Cammell-Laird's practice the length of the stays is adjusted by inserting temporary washers under the nuts at one end, the thickness of which is equal to the estimated expansion of the ball piece taken at the steam pressure and 0·02 inch at least being added to ensure that under steaming conditions there will be a

0·01-inch clearance between the ball pieces and the rings in the bottom of the stuffing-boxes. Whilst the temporary washers are in position the pin holes through the nuts and stays are drilled.

The ball heads are machined to such a form in relation to the cover and the bottom ring of the stuffing-boxes that there is an endwise clearance of 0·03 inch. This ensures that the longitudinal steam load is borne by the stays and not by the cover studs.

It is usual, however, to design these studs so that their total strength is slightly greater than that of the stays, so that in the event of the stays stretching or being incorrectly readjusted, the studs will be strong enough to take the longitudinal load.

It will therefore be noticed that the steam tightness of the joint

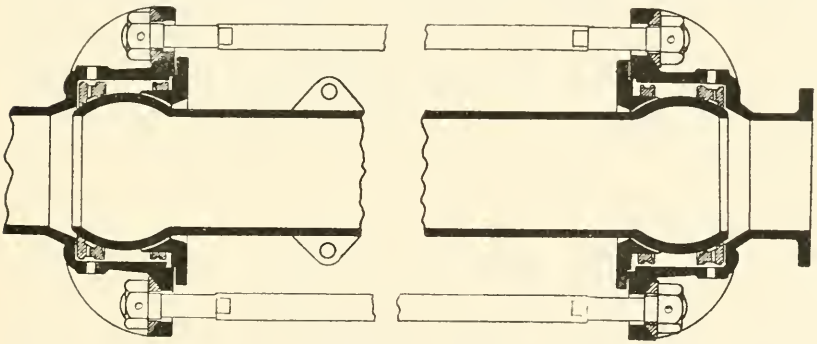


FIG. 297.—Cammell-Laird Ball Expansion Joint.

depends entirely upon the packing, and not on the contact between the ball heads and the adjacent metallic surfaces.

For the ball joints in exhaust steam systems, it is usual to fit steam connections to the stuffing-boxes to prevent air leakage and consequent loss of vacuum.

Intermediate Shut-off Valve.—After passing through the bulkhead valve, the steam enters the cross connecting pipe, and the supply to each engine is governed by the intermediate valve. This valve is of the ordinary screw-down type, and opens against the steam pressure ; consequently, when closed, it is generally steam-tight. A small pass valve and pipe are generally fitted in connection with large valves, to equalise the pressure on opposite sides of the large valve, so that the latter can be opened easily. Gearing is fitted in connection with the valve, to work it from outside the engine room, and it should be accessible without difficulty from the engine room platform.

Bulkhead Valve.—The separate steam pipes leading from the boilers to the engine rooms are controlled by the bulkhead valves, which are of the self-closing type, as shown in Fig. 70. In addition to the bulkhead valve there is generally fitted an emergency shut-off valve of the disc or throttle variety, which is described later.

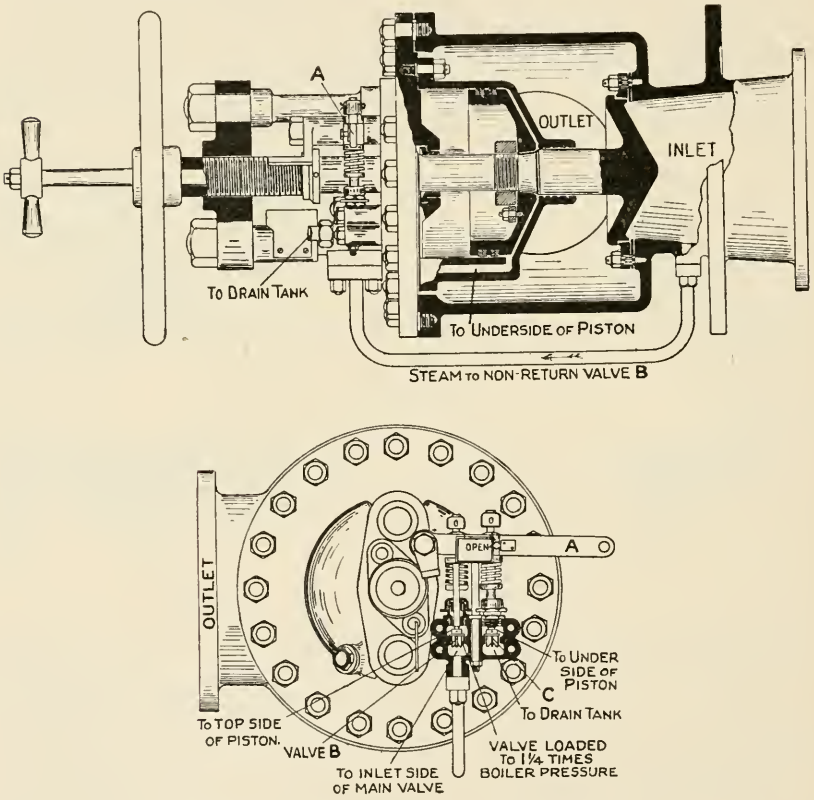


FIG. 298.—Cockburn Bulkhead and Emergency Valve.

The Cockburn-MacNicoll Bulkhead Self-closing and Emergency StopValve (Fig. 298) combines these functions in a satisfactory manner. The valve is fitted on the engine room bulkhead, and a part, as shown, of the valve chest projects through the bulkhead to receive the end of the main steam pipe, which at this position is usually accompanied by an expansion joint.

The main valve is fitted with a balance piston, working inside a

chamber formed within the valve chest. The spindle is solid with the valve and can be used to turn the valve on its seat as in the usual self-closing valve type of fitting. The valve spindle is usually near the horizontal position, and the lift of the valve, or its opening in this case, is regulated by a hand wheel and screw working in an external cross bridge. The valve opens in the first instance from difference of pressure, which is greater on the boiler face, and begins to open when the hand wheel screws the regulating device backwards. As the valve begins to open, the collar on the spindle has an inclined plane action on the LEVER "A," which lifts the spindle of the non-return valve *B* and thus allows steam to pass to the back of the piston and at the same time prevents too sudden opening. As soon as the valve *B* lifts, and before any further opening movement of the main valve takes place, the steam must leak past the piston. The main valve may be closed again by the hand wheel.

The main valve can be opened without external assistance by retaining the emergency gear at SHUT, and opening the hand screw to any desired amount and then returning the lever to its original position of OPEN, when the main valve will gradually open to the desired amount.

Under working conditions, the main valve being open and set for emergency self-closing, the lever is raised and allows the valve *B* to open farther, and at the same time opens the valve *C* which exhausts the steam from in front or main valve side of the piston, which is thus moved forward and closes the main valve. The piston throughout, whether opening or closing be effected, acts as a dashpot, and prevents serious damage from too sudden movement of the main valve, which under high velocity might possibly arise.

This arrangement produces an absolutely steam-tight valve when required, and takes up less space than the two separate fittings known as the bulkhead valve and the emergency valve. Being steam-tight it also does away with another valve, known as the Intermediate Shut-off valve, which was commonly fitted to obtain absolute steam tightness in addition to the other two fittings.

Separator.—The steam in some instances next passes through a separator, the object of which is to extract any water or moisture from the steam before its arrival at the engine.

One form of separator is the *Lancaster central-pipe steam dryer*, shown in Fig. 299. The arrow heads show the course of the steam. Entering through the inlet pipe, the steam is deflected by a web *A*,

made outside the central pipe *B*, and passes spirally downward outside the pipe; thence, turning upwards, it passes through the interior of the pipe to the outlet. Any water deposited on the exterior of the central pipe drains into a cupped lip, *D*, at the lower end of the pipe; and a drain pipe, *E*, leads the water into the bottom of the dryer. Particles of water are also thrown off, by the steam in its passage, by centrifugal action. These are collected by vertical ribs, *C*, made in the inner face of the outer containing vessel, whence the water drains downwards into the bottom of the dryer. The drain water is automatically removed through the connection at *F*, by a steam drain trap. At *G*, a cock or valve is fitted for blowing down and other purposes.

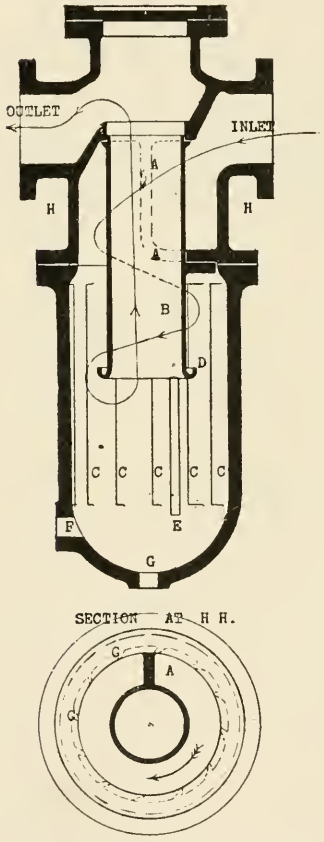


Fig. 299.—Separator or Steam Dryer.

As far as possible, the steam pipes are made *self-draining*. There should be a gradual fall in level from the boiler steam valves to the separator, whence the water can be blown out.

Steam dryers of the type shown in Fig. 72 have now been adapted for pipe steam drainage, and make an efficient system for the combined operations of a separator and a steam trap.

Automatic Separator Blow-Down.—The water separated from the steam during its passage through the separator, and deposited in the bottom of the chamber, is sometimes blown out automatically to the condenser. The automatic apparatus consists of a float rising and falling with the water-level in the separator, a balance weight, and valve, worked by levers attached to the float. The valve is generally of the balanced or double-beat type. The apparatus is preferably contained within the separator, so that the packing otherwise used for the rod and valve spindles is not required. Such packing frequently upsets the perfect balance of the float and weight, producing variable and unreliable

working. A hand blow-through valve is also fitted, and is now usually made of sufficient size for use as a *silent blow-down*. Both the automatic and hand blow-through pipes are connected with the main eduction pipe.

Emergency Valve.—In cases of extreme urgency the intermediate valve cannot usually be closed with sufficient quickness, and in naval vessels a quick-closing valve is fitted between the intermediate valve and the regulating valve. A type of quick-closing valve is shown in Fig. 300, and consists of a nearly circular disc, *A*, which, when turned into required position, either leaves an almost uninterrupted opening, or, as shown in the figure, entirely closes the passage-way. The valve can be worked from the engine-room platform and from outside the engine room, and under ordinary conditions is always wide open. The spindle of the valve withstands the total pressure on the valve

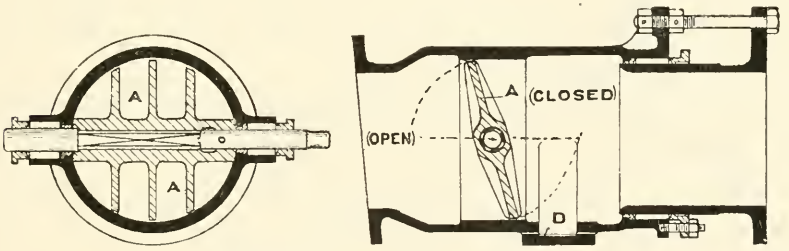


FIG. 300.—Emergency Shut-off Valve.

when closed, and is consequently a substantial fitting; and for the same reason the valve is strengthened by webs, as shown. The emergency fittings saved many lives during the war.

Regulating and Manœuvring Valves.—Steam is admitted to the engine either through the regulating valve; or, with reciprocating engines when only low powers are in use, through a smaller valve, called the manœuvring valve, which is fitted alongside the former, and performs a similar office. The diameter of the smaller valve is generally about one-fourth to one-fifth that of the main regulating valve, and is suitable for powers up to about one-third full power.

In turbine installations the separate nozzle control valves take the place of the manœuvring valve shown for reciprocating engines.

The ordinary valve, with the steam pressing it on to its seating, cannot be opened easily. In modern ships, where the diameter of the main regulating valve is from 15 to 20 inches, the total pressure

on it is many tons, and consequently a balanced valve is now generally used in conjunction with high pressures and large powers. The

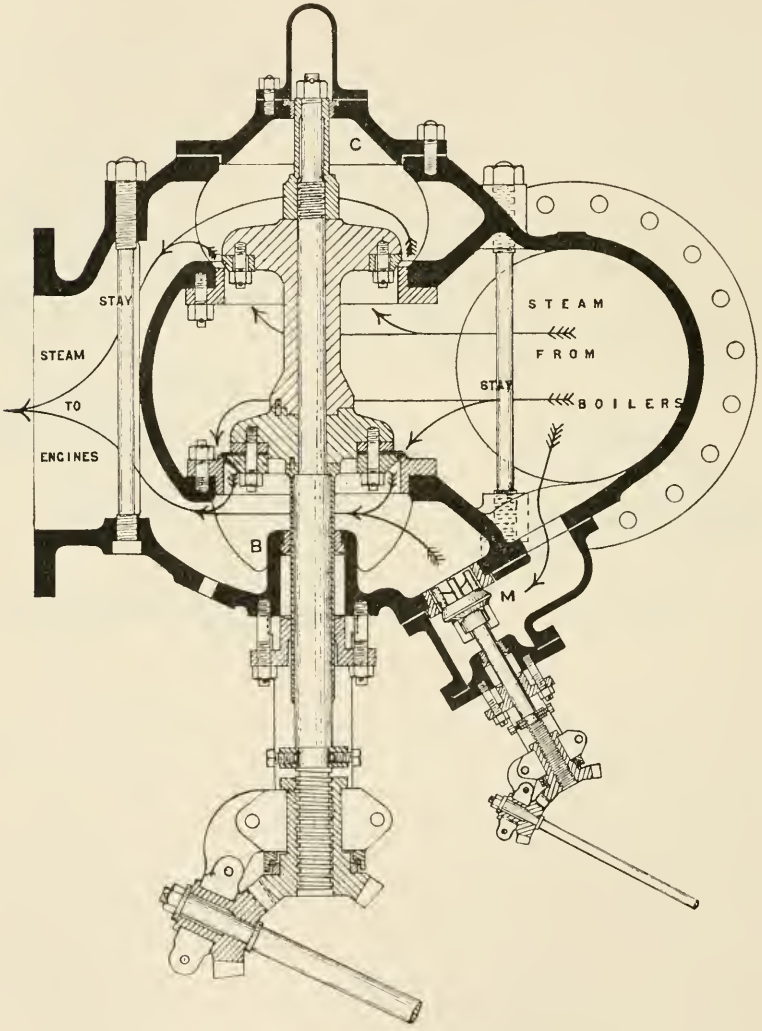


Fig. 301.—Main Steam Regulating and Manœuvring Valves.

balanced valve allows the engines to be handled more easily, and reduces the wear and tear on the threads of the screwed valve spindle.

In Fig. 301 a balanced valve is shown with the steam pressing the

lower valve on to its seating and the inner (upper) valve away from its seating. The two valves being fitted on the same spindle, the end pressure is nearly balanced. The upper valve (that is, the one nearer the cover *C*) is necessarily slightly larger than the lower valve, to allow the latter to be passed into its place ; and there is consequently a slight resultant pressure in the direction of keeping the valve open, when the steam pressure is admitted between the two valves. The spindle is generally packed with asbestos well mixed with black-lead, and in some cases white-metal scrapings are used, and make a working joint. As the spindle is generally below the valve box, it is good practice to raise the stuffing-box *B* inside, and so prevent any water standing above the packing, because it conduces to leakage.

When the valves, the spindle, and the valve box are all made of gun metal, or practically the same material, there is no great difficulty in obtaining tightness under ordinary conditions. For large valve boxes cast steel is now frequently used, but the same material is not suitable for the spindle and valves. There is therefore some variation in the length of the spindle and distance of the valve faces from each other at different temperatures, while the distance of the valve seatings is not necessarily exactly similarly affected. This allows leakage, and special methods have been introduced to meet it, one of which is shown in the figure.

The seatings are removable, and jointed to the casting by studs and nuts, and are generally made of gun metal. Stays are fitted as shown, to resist unbalanced pressures on the walls of the valve box.

The *manœuvring valve M* is shown in the figure ; the pressure enters to the back of the valve, and although of small dimensions, the total pressure on it when closed is very considerable.

Gearing is fitted to the valve spindles if necessary for working from the engine-room platform.

Cockburn Regulating and Emergency Valve.—In addition to the Emergency Valve already illustrated and described in connection with the Bulkhead Valve, which only controls the boiler steam supply, it is necessary with turbines to have another valve for controlling the actual supply of each turbine independently of each other in case of the shaft breaking or the blades stripping or other cause of a sudden nature.

Fig. 302 shows a sectional elevation of the Cockburn arrangement. The main valve is of the balanced type with a flexible disc on the rear valve to effect absolute steam-tightness. The main valve is positively

opened and closed by a hand wheel operated through a double lever

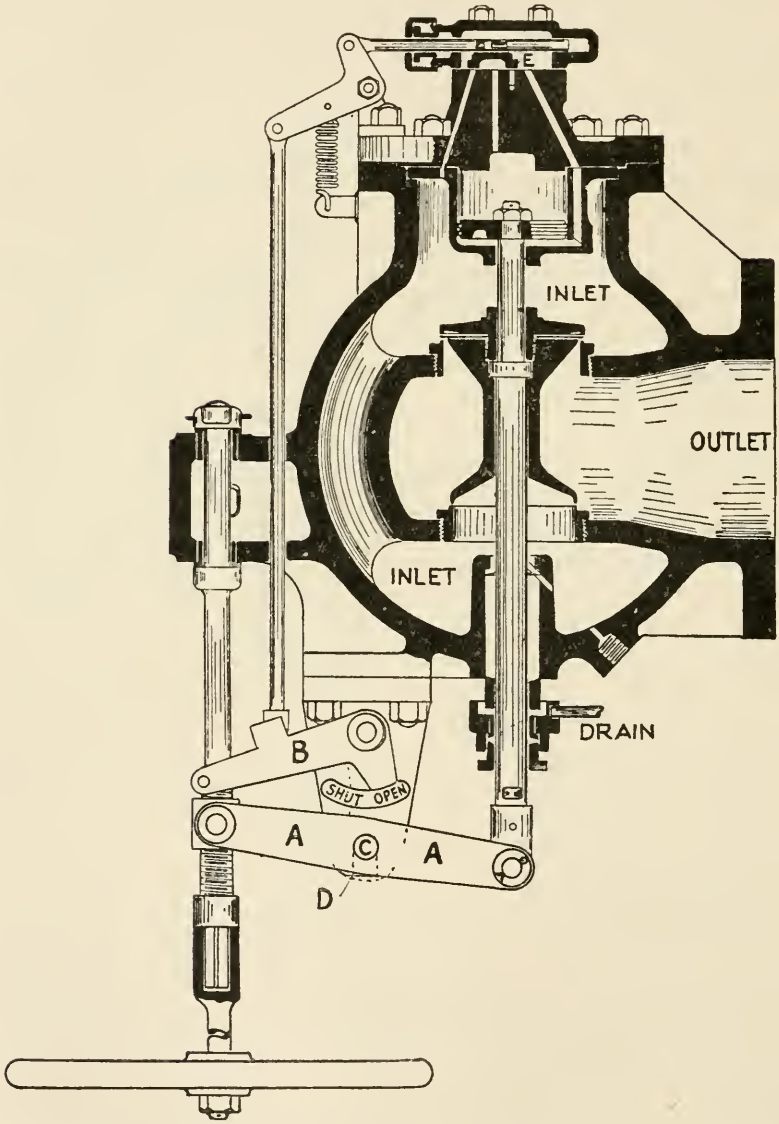


Fig. 302.—Cockburn Regulating and Emergency Valve.

AA, the fulcrum pin *C* of which is either locked by a detent *D* in

connection with the lever *B*, or it may be set free to move a limited distance downwards.

In the position shown, the detent locks the fulcrum of the lever *AA*, and the main regulating valve can be opened by the hand wheel and its opening regulated easily because the control valve *E* is in a neutral position, in which no steam, for working the piston fitted in a chamber above the main valve, is admitted to the control valve chest through the left-hand port from the main inlet, and the exhaust is closed to the condenser.

When the main valve is opened to any extent it can be closed or opened through the emergency gear by moving the lever *B*, which moves the control valve, and, say, opens steam to the control valve chest by uncovering the left-hand port to the space below the piston and thus opens the main valve. Raising *B* still farther the control valve opens the space above the piston to steam and opens exhaust to the space below the piston ; this shuts the main valve. By pulling the lever *B* downwards the main valve can be opened and the detent acted on by the spring returns to the fulcrum pin locking position, and hand wheel operation can be resumed.

The positions of the main regulating and manœuvring valves are shown in Figs. 303, 304, and the other reciprocating engine mountings will now be considered in detail.

Starting Valves.—At certain parts of the stroke of reciprocating engines the admission of steam to the H.P. slide valve will not start the engine, and it is necessary to admit steam to one of the other cylinders to produce motion.

A small D slide valve was originally fitted outside each cylinder for this purpose, and by working a lever, in connection with its slide rod, steam could be admitted to either end of the cylinder as required. Great care was necessary to admit steam to the correct side of the piston for the purpose of starting ; and the lever was generally arranged to incline in the same direction, when steam was admitted, as the connecting rod for *ahead* motion, and it would therefore be crossed with the connecting rod for going *astern*. With very fast-running engines the operation is extremely difficult ; and for all recent types of stage-expansion engines pass valves are fitted, and in some cases in addition to the auxiliary starting valves.

The **bye-pass** starting valve is simply an ordinary stop valve, which can be worked by a lever from the starting platform. One is fitted to each M.P. and one for the L.P. cylinders. Steam can be

Position of Engine Mountings—Plan.

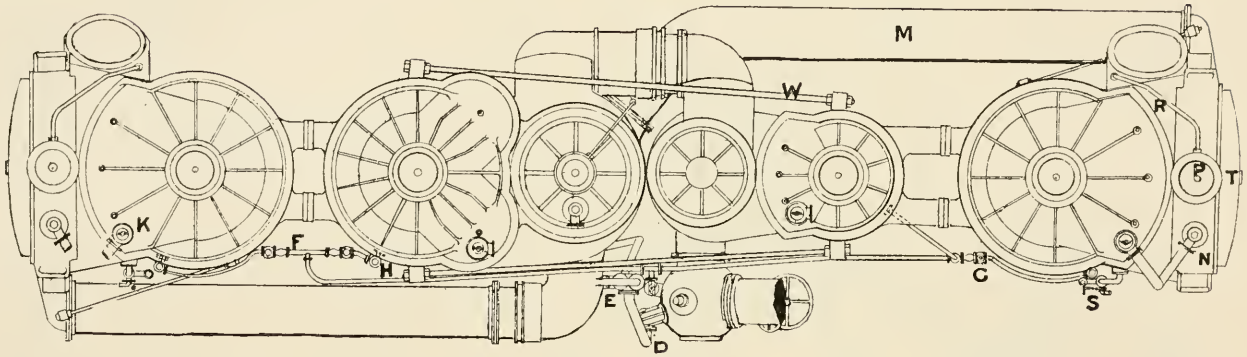
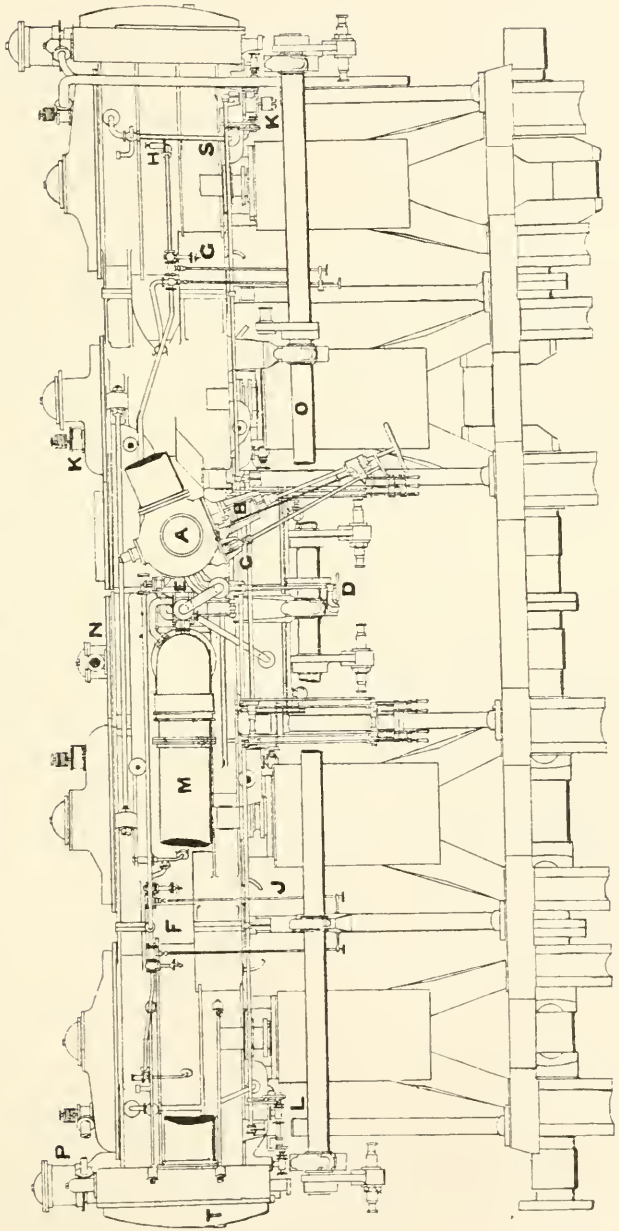


Fig. 303.

A, Main steam regulating valve.
B, Pass valve for *A*.
C, Manœuvring valve.
D, Steam to starting valves.
E, Starting valves and pipes.
F, Steam supply to jackets.
G, Reducing valves for jackets (one on M.P. and each L.P. cylinder).
H, Safety valves for jackets (one on M.P. and each L.P. cylinder).
J, Drain pipes to jackets (one below each cylinder).
K, Cylinder relief valves (one above and below each cylinder).
L, Drain cocks (one below each slide valve and cylinder).
M, Receiver pipes, with expansion joints.
N, Safety valves, in connection with receivers (one above M.P. and each L.P. slide chest).
O, Weigh shaft for reversing gear.
P, Balance cylinders (one above M.P. and each L.P. slide chest).
R, Pipes connecting balance cylinders with eduction and receiver pipes.
S, Indicator pipes and cocks (one for each cylinder).
T, Steam gauge connections for equilibrium rings of L.P. slide valves.
W, Ties or stays between cylinders.



Position of Engine Mountings—Elevation.

Fig. 304.

admitted to the respective receivers, which is then diverted by the slide valve to the correct side of the piston for the required direction of rotation. There is little probability of locking the engines by using the pass valve, although the admission of steam to the receiver may retard the motion of the piston fitted in the next stage *above* it. The piston in the stage below, to which steam is admitted through the slide valve, is larger, and should generally overcome the possible retardation.

In a few particular instances the bye-pass valve is not always suitable. Thus, in a two-cylinder two-stage expansion engine, with the cranks set at right angles, if the H.P. crank is on the dead centre, the L.P. crank is at about half-stroke, and steam will not be admitted through the slide valve by the bye-pass unless the cut-off is later than half-stroke. This is exceptional, but the modern four-cylinder engine is sometimes similarly constructed.

Cylinder Escape Valve.—When, from any cause, an extra pressure or an accumulation of water finds its way into the cylinder, it is allowed to escape through the relief valves fitted on the cylinder above the top end, and below the bottom end, of the piston stroke. In vertical engines the valves are generally placed at top and bottom in places where they do not come in contact with the working liner, and the valve box is fitted with a spigot entering the hole in the cylinder. A *safety valve* is also fitted above the slide chest, to prevent the accumulation of too great pressure. This is particularly necessary for the M.P. and L.P. cylinders, because they are designed to carry a reduced pressure, not the full working pressure, as in the case of the H.P. cylinder.

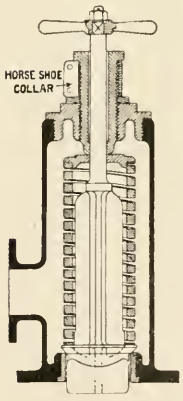


FIG. 305.
Relief Valve.

The escape (or relief) and safety valves are of the same type as a boiler safety valve, and their construction is similar. There is usually a valve in solid connection with its spindle, as shown in Fig. 305, which is fitted with the usual T handle. A spring, fitted to the valve box, and of square section, loads the valve to the required pressure, which is generally a few pounds above the highest working pressure used in the cylinder. The safety valve emits to the upper part of the engine room ; but the relief valves, being connected by a common pipe led down to a safe point below the working platform, generally open into the crank pit.

An accumulation of water in the cylinder is dangerous, as it is practically incompressible; and if the escape is cut off through the slide valve, the relief valve allows it to pass harmlessly into the bilge.

Indicator Pipes.—A large pipe, about 2 inches in diameter for a large engine, is fitted to each cylinder, and is connected with the spaces above and below the piston stroke. At the upper, or any convenient part, a two-way asbestos-packed cock is fitted in the pipe, and has a projection fitted in connection with it for use with the indicator. By means of the cock, connection can be opened with either end of the cylinder as required for the indicator to take readings. A system of levers is fitted from the piston-rod end for working the indicator in unison with it, and one of these levers is generally used for working the engine revolution counter.

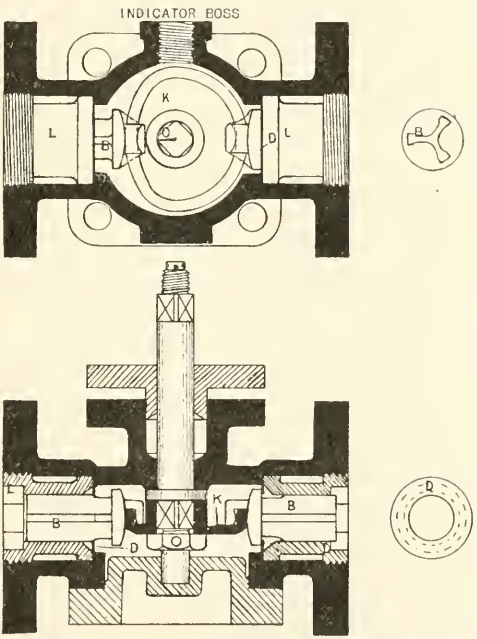


FIG. 306.—Bevis-Gibson Indicator Pipe Valve.

In Fig. 306 a type of valve for use with the indicator is shown. The seating is flexible, and made of thin sheet-metal, in the shape of a ring, as at *D*. The valve is of ordinary construction, with three webs, as at *B*, with a narrow flat seating; above the valve a lug is made, catching over the lip of the cam *K*, which moves the valve through its means. The cam is formed with a spindle, which is worked from outside the valve box, as in the case of an ordinary asbestos-packed cock, and when one valve is closed the other may be either closed or open. A mark on top of the spindle shows the direction in which the opening is pointed. The seating is fixed in position by a valve case *L* which screws into the valve box. The valves can be inserted or removed either by removing the bottom cover or by taking down the valve box and removing the

seating. The pressure keeps the flexible seating in tight contact with the valve, and thus obtains a steam-tight joint. A screwed hole is fitted on top of the valve box for inserting the indicator.

A similar arrangement is used for other purposes, as a multiple way valve, and it is then made with a flexible valve as well as seating. The flexible valves are Bevis and Gibson's patent.

Jacket Steam.—The steam for the cylinder jackets is generally taken from the steam side of the main regulating valve, and a shut-off valve is fitted at this end of the pipe to control the supply to all the jackets. A branch is led from this supply pipe to each cylinder jacket, and each of the supplies to the M.P. and L.P. jacket is controlled by a separate stop valve, a reducing valve (to keep the pressure below that for which the cylinder is fitted and made), and a safety valve (which gives warning to the engine room, should the reducing valve fail to act).

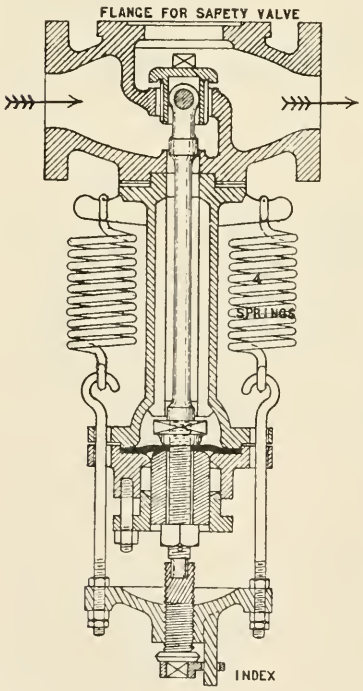


Fig. 307.—Reducing Valve.

(The steam for the starting valves is also frequently led from the same pipe that supplies the jackets.)

A separate supply is fitted for the H.P. cylinder jacket of large engines, and is led from the slide chest independently to the jacket space. The pressure and temperature in this way cannot exceed that due to the maximum pressure in the cylinder, which, if excessive, frequently causes undue friction of the working parts. The control valve for this supply must be non-return or the pressure from the other supply to the jacket might escape into the slide chest.

Reducing Valve.—A type of reducing valve suited for the jacket steam is shown in Fig. 307. The springs are in tension, and tend to keep the valve open at all times. When the pressure on the reduced-pressure side of the valve exceeds that for which the springs are set,

the valve closes, and prevents the pressure rising. The inlet pressure tends to keep the valve open, and this pressure is balanced by the pressure in an opposite direction on an india-rubber ring, shown in black section, and consequently the opening or closing of the valve depends on the springs and the pressure on the reduced side of the valve. The tension on the springs can be regulated to a moderate extent by a screw fitting over the lower end of the valve spindle, the movement and pressure being shown by an index.

Steam Drain System.—The drainage of the steam and exhaust systems is much simplified by sloping the main run of pipes gradually, without pockets or dips, from the source of supply to some convenient depository, such as the separator and condenser, and this is usually done. Separators are sometimes fitted to the main steam pipes, as already described, and to the auxiliary steam and exhaust pipes at their lowest parts in each engine room. In addition, drains are fitted to the valve boxes connected with these pipes, and led, through drain traps, into the steam drain pipe common to the machinery and boiler spaces.

The water in this steam drain pipe gravitates into a tank, fitted in the engine room or boiler room as convenient, whence it is removed by a pump, which delivers into the grease extractor or feed tank as required. The tank is closed and fitted with a relief valve, manhole, and cover and drain cock, and a float and index, to show the quantity of water in it. The drain water runs into branches connected with this drain pipe at various parts of its run.

The safety-valve boxes are drained into open-topped drip tanks, which overflow into the drain pipe, and so to the drain tank. The steam cylinder and slide-valve chests of the auxiliary engines are drained into the bilge, and not into the drain system. By an arrangement of one or two of the branches to the auxiliary engines from the steam pipes at various parts of the ship, a fall can be made, which serves as a drain for the general run of pipe ; and the drain pipe from this branch, through a drain trap and into the tank, relieves the pipe of all accumulations of water.

Jacket Drains.—In some ships each cylinder-jacket drain pipe is connected with a separate vertical cylindrical gun-metal tank, generally fixed to one of the columns supporting the cylinder. A water gauge is fitted to the tank, and any water accumulating can be blown out as required into the main eduction pipe. If practicable, however, this drain water should be used to warm the feed water, and thus

promote some economy. The heating of the feed water to the moderate temperature which is obtained by the drain water, does not reduce the efficacy of the pumps if the air and vapour expelled by the effect of heat is allowed to escape. The drain should be fitted with a nozzle inside the tank, to effect the heating quietly, without shock, and in a definite direction, for circulating the water inside the tank (see Chapter XXV.).

An unnecessary loss of heat can be prevented by allowing the condensed water to accumulate in the cylindrical tank, and keeping this at a regular working level by adjusting the drain cock. As only water is then blown out, very little heat is wasted. In the mercantile marine, and generally on shore, the jacket drains are led through steam drain traps, which automatically serve the above purpose.

Cylinder Drains.—When raising steam, or the engines are standing under steam, there is some condensation in the cylinders, and the water is not swept out by the exhaust, which generally happens when the engines are moving. The drains from each of the H.P. and M.P. cylinders and slide-valve chests are fitted with a cock close to and below the cylinder, which can be worked from the starting platform. These cocks are in connection with separate pipes, leading to switch cocks, which can be connected with either the main condenser or the bilge. The leads into the condenser are fitted with shut-off valves at the discharge end, and the leads into the bilge are cut off at a convenient height, so as to be visible from the starting platform.

The same arrangement is fitted for the L.P. cylinder and slide-valve chest drains, but the discharge is conducted into the steam side of the condensers and the bilge. The lead to the condenser is inclined downwards throughout its entire length, and a shut-off valve is fitted on the condenser end. The lead to the bilge is fitted with a non-return valve, near the switch cock, and serves to test the presence of water, or otherwise, in the cylinder (see Chapter XXV.).

Turbine Manœuvring Valve.—For turbine engines a special arrangement is required for regulating and reversing the several shafts, and in the *Mauretania* the system shown in Fig. 308 has been adopted for each of the middle shafts in connection with the L.P. Ahead and Astern turbines. The valves, one for each turbine, are worked by means of wheels on the starting platform. The wheel operates the valve of a Brown's engine (Fig. 209 type), which opens either valve

as required without opening the other. When the Brown piston is in mid-stroke both valves are closed ; the forward stroke opens the astern turbine valve and the return stroke opens the L.P. Ahead turbine valve. The upper of the double-beat valves is of the Bevis-Gibson type and the lower is of the piston type ; and in the mid position of the actuating gear there is a slight tendency, from the inlet pressure and the larger size of the upper valve, for the steam to be shut, and this action is assisted by a spring shown. When the Brown engine

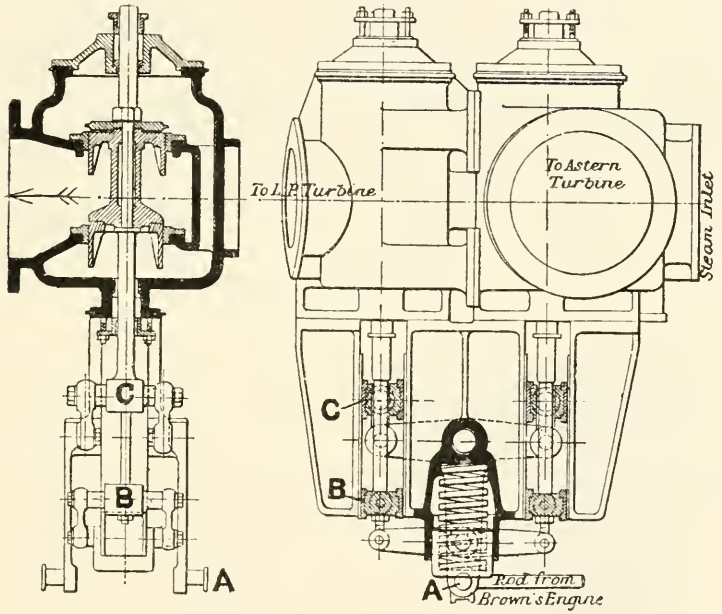


FIG. 308.—Manœuvring Valves (*Mauretania*).

pushes the end *A* of the *T* lever to the left, the astern valve remains closed and steam is admitted to the L.P. Ahead turbine. Conversely, if *A* is moved to the right the L.P. Ahead valve remains closed, and steam is admitted to the astern turbine. In the *Mauretania* a hand pump is provided at the starting pedestal, which enables the manœuvring valves to be opened and closed. Water pressure from the hand pump thus can operate the Brown engine, in the event of the latter breaking down. The manœuvring valves are used only for entering and leaving harbour, etc., and at other times the steam is shut off the steam inlet to them.

In Fig. 308A the arrangement fitted by Brown-Boveri & Co. in some turbine mercantile steamers is shown with a reference from which the general idea of control of the turbines is regulated, and also

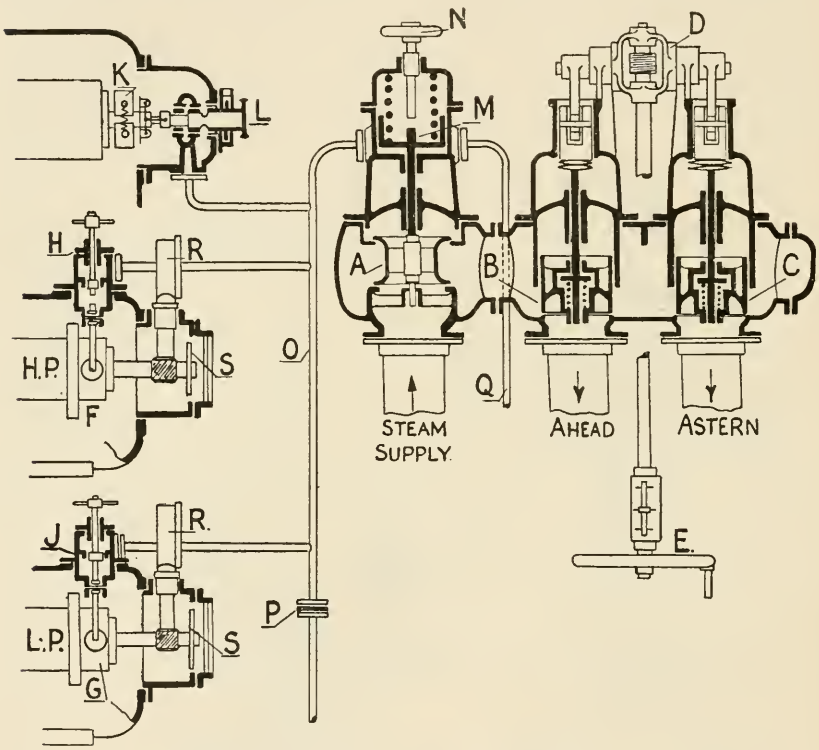


FIG. 308A.—Brown-Boveri Turbine Control System.

A, Safety valve.
B, Ahead admission valve.
C, Astern admission valve.
D, Valve control gear.
E, Hand control wheel.
F, Overspeed governor, H.P.
G, Overspeed governor, L.P.
H, Trigger gear, H.P. governor.
J, Trigger gear, L.P. governor.
K, Speed regulating governor.
L, Speed control.
M, Servo-motor piston.
N, Hand wheel.
O, Oil pressure pipe.
P, Diaphragm.
Q, Oil return pipe.
R, Tachometer.
S, Direction of rotation indicators.

the system of overload or overspeed governors is shown. This system of control is hand operated by the wheel *E*.

Balanced Nozzle Control Valve.—This valve was introduced by Messrs. Cammell-Laird and is made by Messrs. Cockburns. The valve is of the hollow cylindrical type with two seatings, one of which

is of the flexible metallic disc type. The pressure being thus balanced, the valve can be easily closed and opened; opening a large valve against a pressure is sometimes extremely difficult, and leads to jambing and other troubles in starting and stopping. By using metals in the construction which have similar coefficients of expansion, the valve

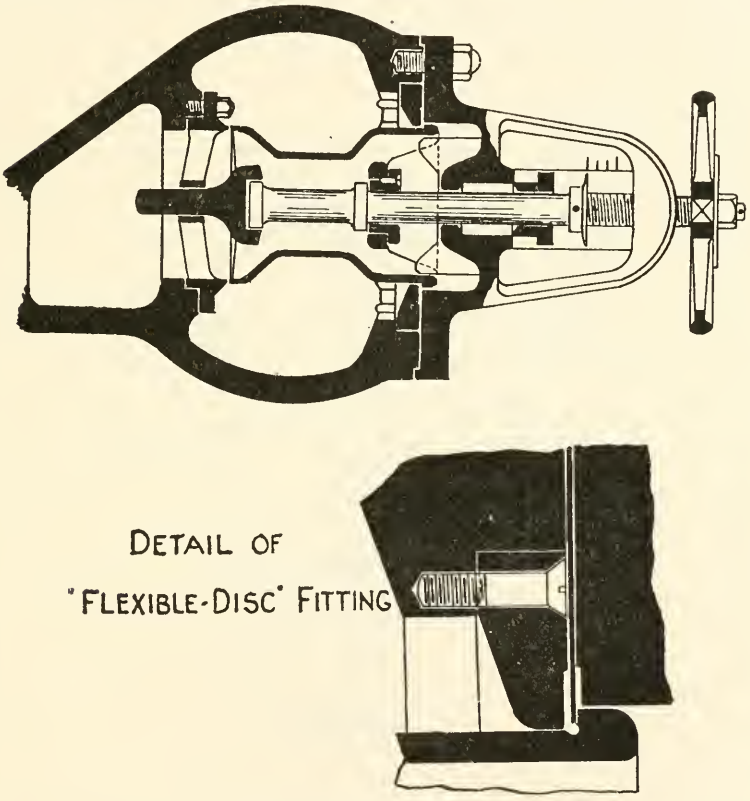


FIG. 309.—Nozzle Control Valve.

is efficiently tight on its seatings, and any slight difference which may arise is compensated by the flexible seating system.

When open, practically all the steam will pass through the inner opening, and therefore due allowance is made at this point for the full opening necessary, irrespective of the disc opening. The wear or scoring of the disc seating is also for the same reason very small and almost negligible.

Cockburn R.D.S. Steam Trap.—This arrangement is shown in Fig. 310, and consists of an outer casing with a central pipe leading downwards. A float nearly fills the chamber and is fitted with a central pipe which serves as a vertical guide for vertical movement on the fixed central pipe attached to the cover of the chamber. A needle-pointed valve is fitted at the lower end of the guide and fits into a contingent hole in the fixed pipe.

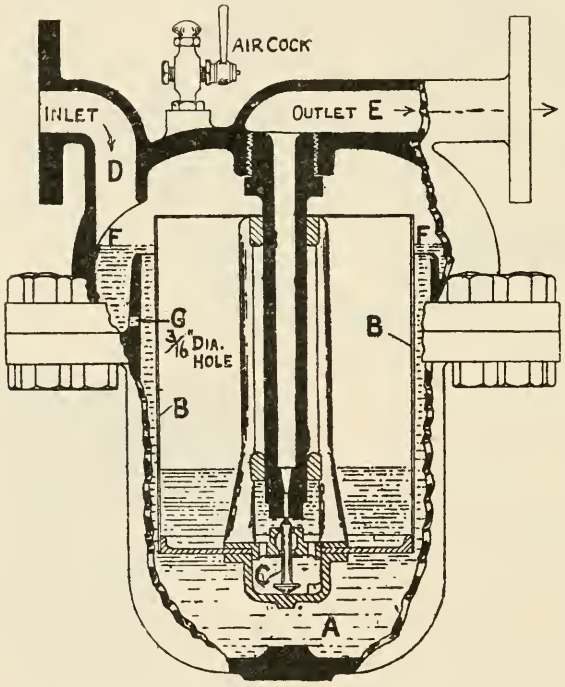


Fig. 310.—Cockburn R.D.S. Steam Drain Trap.

When water gets into the float in sufficient quantity to upset the balance of flotation, the float lowers and opens the valve, and the pressure drives the water out until the float begins to rise and close the valve. Quick movement is now necessary to prevent dribbling, and possibly steam leak; this movement is obtained by supplying water from the reservoir through the hole *G* to make up the lost displacement of the float. The arrangement is thus automatic and requires no adjustment for pressure or temperature.

An *expansion drain trap* works on the principle of different rates

of expansion of two component parts. Academically, the temperatures of steam and water in the same vessel and under the same pressure are the same ; an expansion drain trap would not work if this were always absolutely true. Actually, any water contained in the apparatus cools more rapidly, due to its greater conductivity, than the steam from which it is condensed, and therefore this cooling produces a contraction of a tube containing water, and opens a valve.

CHAPTER XXIV

CONDENSER, AIR PUMP, AND CIRCULATING PUMP

Introductory.—One pint of water makes a *relative volume* of 1643 pints of steam under ordinary atmospheric pressure, or 11,000 pints under a pressure of 2 lb. absolute, or 20,700 pints under a pressure of 1 lb. absolute, which is equivalent to about 28 inches of vacuum by the gauge.

Therefore if 20,000 pints of steam be suddenly and entirely condensed in a condenser where a vacuum of about 28 inches is obtainable, they would fill only about one pint as water ; and the remaining space or volume contains practically no vapour, with a consequent very low pressure or tension in the space of 19,999 pints. Steam is a readily expanding fluid, and the exhaust steam from the engine is not all and entirely condensed in the cooling chamber ; a very small quantity expands as the pressure falls, and, expanding as vapour, fills the space left by the condensation of the remainder. A small proportion of air and uncondensable vapour also enters with the steam ; this prevents the formation of a perfect vacuum, as well as some purely mechanical causes.

The reduction of the exhaust pressure by using a condenser decreases the back, and therefore retarding, pressure on the *LP* piston, and thus increases the mean pressure and power obtainable from the engine, without requiring an increased quantity of steam from the boiler. Economy of steam for an equal power is thus obtained by using a condenser, and a further economy is also obtainable from the possibility of using a greater ratio of expansion with a decreased back pressure.

Jet Condenser.—With a jet condenser the steam is condensed by mixing the cooling water, in the form of spray, with the steam. For each 1 lb. by weight of steam condensed about 40 to 60 lb. of cooling water are required ; and if the cooling water is taken from the

sea, the mixture produced is thus practically sea-water. This mixture when used as feed water leaves a lime scale on the heating surfaces of the boilers, and reduces their efficiency. An accumulation of scale is dangerous in high-pressure boilers, and therefore for marine purposes the jet condenser has been generally and almost entirely abandoned.

Surface Condenser. — For sea-going steamers, and for land engines wherever possible, surface condensation, in which the cooling water and condensed steam remain separated, is almost exclusively used. In addition to the advantage of the jet condenser in reducing the back pressure on the L.P. piston (with a contingent increase of expansion), all the condensed water—except a small percentage of loss from leakage—is saved and returned to the boiler as feed water, uncontaminated by sea-water, by using a surface condenser.

In Fig. 311, *H* is the condenser body, at each end of which is a flat tube plate jointed to the body by flanges formed on the outer casing. The bolts and nuts used for these joints also serve for the joints of the end covers. The tube plates are generally about 1 inch thick, thinned down to about $\frac{3}{4}$-inch at the joining surfaces. Holes are drilled through them, and a great number of small pipes, or tubes, are fitted from tube plate to tube plate and made tight at their ends in the holes by packed glands. The internal passages through the tubes connect the two end spaces, between the tube plates and the end covers, with each other. The surfaces of these tubes constitute the cooling surface of the condenser, and a certain surface area is provided in the condenser according to the weight of steam which is required to be condensed per hour for the total horse-power.

The cooling water enters from the sea by the main inlet *D* and sea suction valves, passes through a pipe, *E*, into the circulating pump, *F*, and is driven by the pump through the pipe, *G*, into the space at the end of the condenser between the tube place and cover. Thence it passes internally through the tubes to the other end of the condenser ; and, in the arrangement shown, back through the upper rows of tubes to the front end, and overboard through the pipe *V* and the outlet valve. The direction of circulation is shown by the arrows. The water space at the front end is divided horizontally, so that the water cannot take a short cut overboard ; and in this plate a valve may be fitted, which can be opened when the circulating pump is used to pump out the bilge—so that the dirty water need not pass through the tubes—and to increase the efficiency of the pump. (The division

plate is sometimes fitted vertically, when the inlet and outlet are fitted near each other.)

The *covers* should be shaped at the inlet and outlet openings to deflect the water equally over the tube plate, so that each tube should do its share of the cooling ; the covers are now generally adapted for

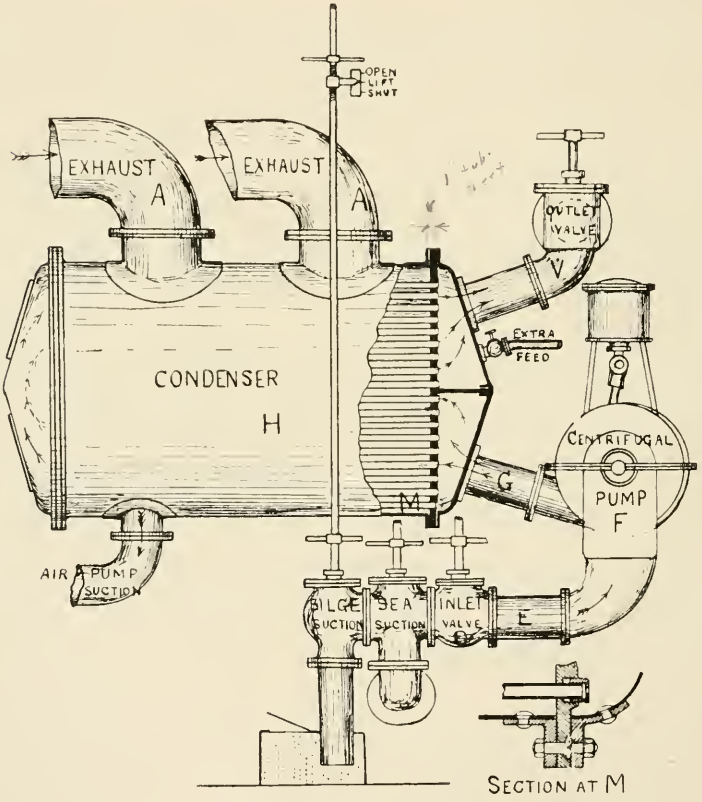


Fig. 311.—Surface Condenser.

this purpose. Small handholes, about 10 to 12 inches in diameter, are fitted in the covers for facilitating examination of, and small repairs to, the tube ends. The covers are usually made with webs, to give them sufficient stiffness and strength. The tube-plate joint is shown in Fig. 312, and it should be noticed that the cover can be removed without disturbing the joint between the body of the condenser and the tube plate.

In a few cases the steam passes through the tubes internally, while the cooling water is circulated externally to the tubes. This arrangement has one advantage, in that leakage at the tube ends can be more readily detected, and the test pressure is applied in the same direction as when the condenser is at work. There are some disadvantages, however, generally of a practical nature, and in the majority of cases the condensers are arranged, as fully described and illustrated in this chapter, with water inside the tubes.

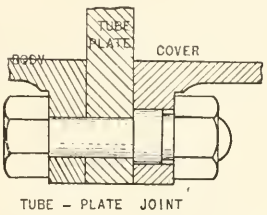


FIG. 312.—Tube-plate Joint.

A direct tube arrangement is sometimes fitted, particularly in small vessels, when the water passes through the tubes in one direction only and then directly overboard ; a division plate is then unnecessary.

Path of the Exhaust Steam.—Referring to Fig. 314, which shows a Contraflo type, the steam enters the condenser from the eduction pipe at the top of the casing ; and, dribbling down externally to the tubes, becomes condensed on the cooled surface of the tubes during its passage to the bottom, whence it is removed by the wet and dry air pumps. The connections and mountings fitted to the main condenser on the steam side are :

At the top—

1. Vacuum, and combined pressure and vacuum gauges.
2. Eduction pipe from main engine.
3. Exhaust from reversing engine, if fitted, and one main circulating engine.
4. Silent blow-off.

(Numbers 3 and 4 are usually connected with the eduction pipe.)

At the bottom—

1. Main air-pump suctions ; air suction and condensate suction.
2. Auxiliary air-pump suction, or hot-well pump suction, if fitted.

If independent air pumps are fitted, neither of these arrangements is required.

3. Connection with the reserve tanks, to draw water from them to *make up* the feed water loss.
4. Drain from L.P. cylinder fitted with a baffle, as shown in Fig. 313. If this drain is led into the crank pit, bilge, or feed tank, water may be drawn up *into* the cylinder and damage done.

The connections and mountings fitted to the auxiliary condenser are :

At the top—

1. Combined pressure and vacuum gauge; sometimes two are fitted.

2. Auxiliary exhaust pipe.

3. Exhaust from the auxiliary air pump and one main circulating engine.

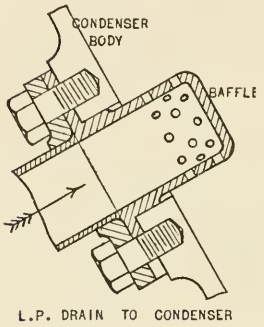


Fig. 313.—Inlet Baffle.

4. Pipe conveying the *gained steam* from the evaporator, when used for boiler *make-up*.

5. Exhaust from steering and dynamo engines. Separate pipes are generally fitted for these engines to overcome the tendency to fluctuation of speed from a varying exhaust pressure.

At the bottom—

1. Air-pump suction.

2. Drains from the H.P. and M.P. cylinders and slide-valve chests fitted with baffles, as shown in Fig. 313; by means of switch cocks these drains can be led into the bilge.

3. Drains from the main engine cylinder jackets. In some cases these drains are led into the feed tank or hot well, and some economy is effected by heating the feed water.

4. Drain from the *primary* steam of the evaporator coils. In some cases the connection is made with the top of the condenser.

(When no auxiliary condenser is fitted, the connections are generally made with the main condenser, through the eduction pipe.)

Baffles.—All inlets to the steam side of the condenser are covered by baffle plates, secured to the casing, to absorb the momentum of the entering steam, and thus prevent any violent or direct impingement on the tubes, which may produce vibration, and consequent abrasion or splitting. The baffle plate for the main eduction pipe is strongly held to the casing, and perforated.

Condenser Tubes.—For naval purposes the tubes are made of gun metal of special composition, and tested to an internal pressure of 300 lb. per square inch. The composition generally adopted consists of at least 70 per cent copper, 1 per cent tin, and the remainder zinc. The casing and all parts of the condensers are made of gun metal of as nearly as possible the same composition as the tubes, with not less than 1 per cent of tin.

For naval purposes the tubes are $\frac{5}{8}$-inch diameter externally, $\frac{1}{20}$-inch thick, and the length is now standardised to some multiple of 5 feet. They are pitched at the corners of an equilateral triangle, at a distance of about 1 inch from centre to centre.

For the mercantile marine the tubes are sometimes made of pure copper, and are generally $\frac{3}{4}$-inch in diameter.

The surface area of a tube $\frac{5}{8}$-inch diameter and 5 feet long is about 0·818 square foot ; and about 0·5 to 0·55 square feet of cooling surface

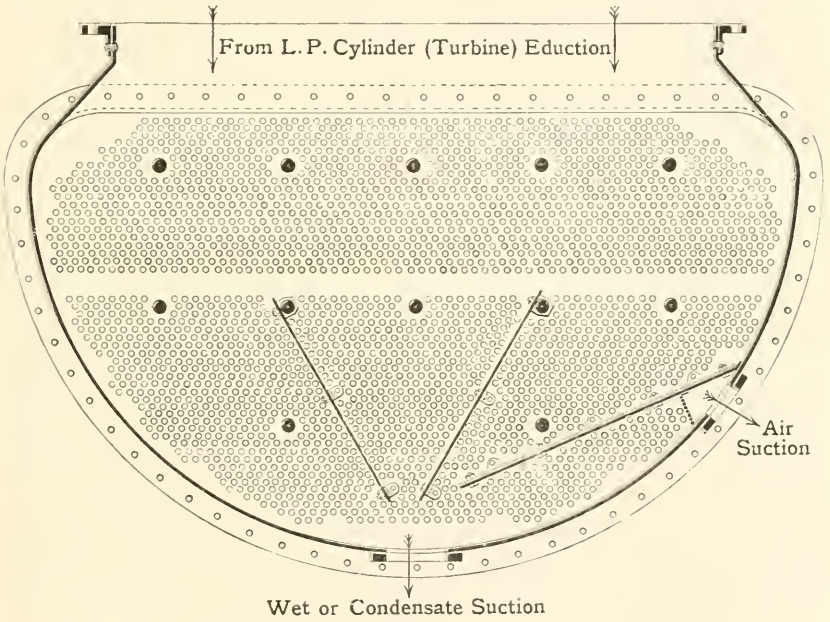


FIG. 314.—Contraflo Condenser.

being allowed per S.H.P., the number of tubes and glands is very great.

Tube Glands.—The tubes are passed through holes in the tube plates, and when in place the ends are about level with the outer faces of the plates. A screwed gland is inserted into the plate over the end of each tube, as shown in Fig. 315, and a cotton grommet under the gland makes a tight joint with the tube plate. The glands are made with an inner flange, so that the tube cannot work through them, and about $\frac{1}{16}$-inch clearance is allowed for unequal expansion between the end of the tube and the flange. A slot is cut across the

end of each gland for use when screwing up or removing. The glands are about $1\frac{3}{16}$-inch diameter externally for a $\frac{5}{8}$-inch tube.

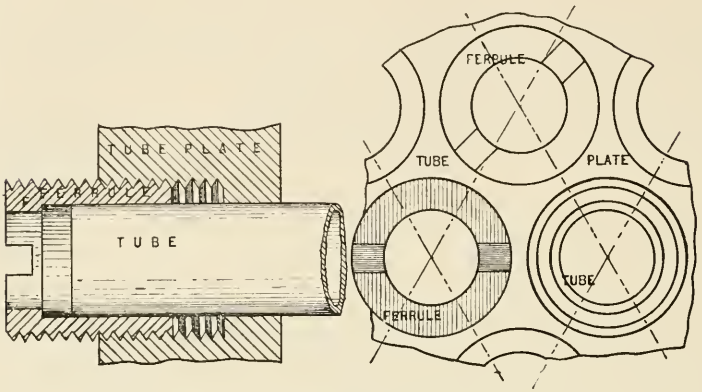


Fig. 315.—Tube Packing Gland.

Condenser Stays.—The tubes do not support the tube plates, they merely form the connection between the outer sides of the plates for the water to pass through, the tubes forming the cooling surface, without mixing with the condensed steam. The outer, or circulating water, side of each plate is subject to a pressure, or head of water, due to the depth at which the condenser is placed below the water-level outside the ship; the inner side of the plate is subject to less than atmospheric pressure in proportion to the vacuum, and thus the tendency to collapse is augmented on the plates, and a total pressure of about 30 lb. per square inch must be provided for. Stays are therefore fitted from plate to plate, at proper intervals, and in

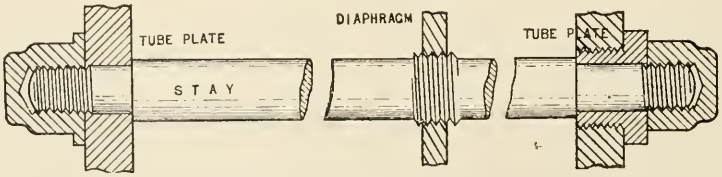


Fig. 316.—Condenser Stays.

the place of some tubes which are consequently omitted. A form of stay, which can be removed without disturbing the tubes, is shown in Fig. 316; the hole in the front plate is large enough to allow the withdrawal of the shoulder fitted to support the plate at each end.

In many cases the stays were simply fitted with check nuts on opposite sides of the plate, but these could not usually be removed or set up without removing the tubes in the vicinity.

If the condenser is not circular in shape, and some ground space is saved by making it flat-sided, the flat sides must be prevented from bulging. Stays are therefore fitted across the condenser body ; and they are conveniently made of the form described above, with bosses on the casing, to give the necessary local strength for the nut and screw thread.

Diaphragm Plate.—When the condenser tubes are over 5 feet in length, it is necessary to support them, so that the vibration and general sagging do not bring them in contact with each other. Supporting plates, or diaphragms, are therefore fitted not more than 30 inches from each other, or from the tube plates, through which the whole of the tubes are reeved. The necessary holes in the diaphragm plate should be a moderately good fit for the tubes passing through them, or the vibration within the holes causes abrasion. In some cases india-rubber rings are fitted over the tubes, to absorb the vibration and to keep the tubes properly apart. If the diaphragm plates are fitted exactly at equal distances the vibrations tend to synchronise, and increase in intensity ; by fitting the plates at unequal distances the vibrations tend to decrease in intensity. In some cases the tubes are about 10 feet long, and the distances between the diaphragm and tube plates are $28\frac{3}{4}$, $28\frac{1}{2}$, 28, and 30 inches respectively.

A **weed trap** is fitted to nearly all inlet pipes for the circulating water where the inlet is not very deeply immersed, as in destroyers and for auxiliary condensers. It is generally fitted at *E* (Fig. 311), so that, by removing a cover, any weed that may have been caught can be removed, and in most cases the grating can be taken out and cleaned. The inlet and outlet valves should be closed when clearing the weed trap, and of course the steam must not be exhausted into the condenser while the water is shut off. Sometimes the direction of flow of the circulating water can be reversed, and the weed thus cleared from the grating and tubes ; or a fire-hose can be connected with the pump to effect the same purpose.

Protecting plates are fitted in the water spaces at the ends of the condenser, in electrical connection with the tube plates. The plates were made of zinc, then cast iron was tried, and they are now almost invariably made of mild steel. Protection plates are also fitted in the bottom part of the condenser on the steam side and to the weed-

trap grating, when one is fitted. If the plates are secured to the handhole covers—until recently a common practice—care must be taken that they are not insulated from the tube plates by the jointing material. A modern method of fitting steel protectors is shown in Fig. 317. It secures electrical contact with the tube plates at all times.

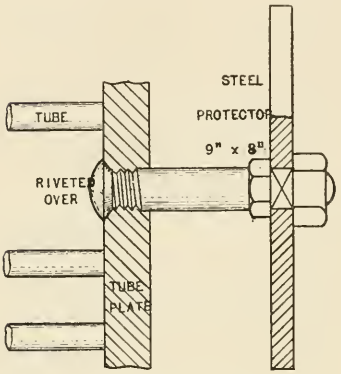


Fig. 317.—Steel Protecting Plate.

The *failure of condensers* is generally due to leakage from either mechanical or corrosive causes. The mechanical causes are generally: (1) inefficient baffling, with a too contracted escape from below the baffle and a restricted flow from the eduction pipe; (2) inefficient packing of the tube glands, or shrinkage of the packing when dry—for this reason the water should pass through the tubes, and the condenser should not be drained unnecessarily on the circulating side; and (3) too great clearance in the holes in the diaphragm plate.

If a sufficiency of lime is mixed with the feed water, the effect of acid corrosion is practically overcome. The other corrosive agent is the air or oxygen set free from the condensed steam or water, which in its nascent state has a great affinity for metals. If the circulating inlet is below the condenser, and the discharge gradually rises to the outlet, the heating of the circulating water, obtained by the condensation of the steam, evolves the oxygen and air, which pass away through the outlet, and little corrosion takes place; consequently, when fitted in this way, after shutting off steam, the inlet should be closed to prevent the entry of air, and the outlet left open to allow any air to escape. It is now customary to fit air outlet pipes near the highest points of the circulating water spaces to allow the air to escape; this practice is of undoubted advantage in preventing corrosion. On the steam side the same reasoning holds good; but as the air may have access through the eduction pipe as well as the air-pump suction, these should be kept closed when the condenser is not in use, and no air allowed to enter. It is impossible to drain the condenser on the steam side without leaving moisture on the tube surfaces, and the exclusion of air is the best practice.

Air Pump.—Originally the cylinder itself was cooled after each

stroke, to produce a vacuum; but it was soon evident that a separate vessel for this purpose would be advantageous, and James Watt introduced the condenser. In this way the steam cylinder is always kept as hot as possible, and the separate chamber in connection with the cylinder, called the *condenser*, is kept as cold as possible. Hence it follows that directly the steam enters the cylinder it is ready to act more effectively; and directly communication is opened with the condenser the exhaust steam makes its escape, assisting in its own withdrawal by producing a partial vacuum from condensation in the condenser. At this point another difficulty arises—the condensed water and uncondensable air and gases would soon fill the condenser, unless continuously removed.

The air pump is used for this purpose, and its function is therefore to pump the condensed steam and uncondensable vapours and gases from the condenser, and at the same time to decrease the pressure in the condenser by continually pumping out its contents. The condenser and air pump are mutually dependent on each other for reducing the temperature and pressure of the exhaust respectively. The effect of this reduction of temperature and pressure is to reduce the back pressure on the piston (only the L.P. piston directly in a stage-expansion engine), and also to increase the possible ratio of expansion, thus producing economy of steam and fuel.

The air pump discharges its contents into either the hot well, the grease filter, or the feed tank; from one or all of these the air is allowed to escape through vent pipes fitted for the purpose. From the feed tank the boiler feed pumps take their suction, and thus the same water is used over and over again. With a surface condenser, and where the packing glands of the cylinders and rods are well fitted, the amount of air which actually enters the condenser is small, and thus the air pump could be of small size; but provision must be made for emergencies, and, to prevent overcharging, the air pump is generally a large fitting.

In mercantile practice the air pump was usually a single-acting bucket pump, generally worked off the main engines either by levers from the piston-rod end or by direct connection with one of the pistons; the former method is preferable. Double-acting pumps are now frequently used, and save space and weight. Recently, for naval purposes, and also for some of the better class of mercantile marine, the air pump is worked by an independent engine. Two air pumps are now fitted for each main engine in the larger vessels, each of

which, being independent of the other, can be worked or repaired as required.

In Fig. 318 a vertical air pump is shown. There are in this single-acting type three sets of valves, one each for *delivery*, *bucket*, and *suction*.

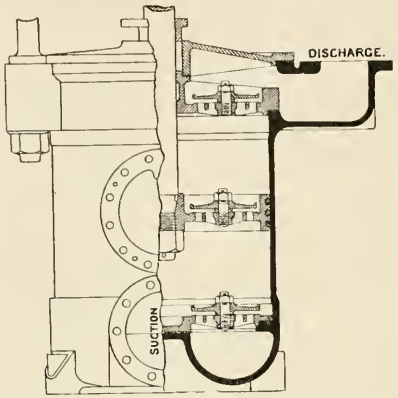


Fig. 318.—Single-acting Air Pump.

The valves are preferably made of gun metal; but india-rubber is sometimes used, and it is then specially prepared and vulcanised. The bucket, or plunger, is fitted with packing, to make it a working fit in the barrel, and for large engines a junk ring is fitted to it for fitting or renewing the packing. The packing is preferably made up of two or more vulcanite rings, generally uncut or solid. Hemp is also used, and for small engines is efficient; and in some cases a broad ring, made of gun metal, is used, fitted with a tongue piece and water grooves. Solid plungers are occasionally used for air pumps, and the fitting of water grooves adds considerably to their efficiency.

Metallic Valves.—A type of valve, known as Kinghorn's, is very frequently fitted to air pumps, and is generally most efficient and durable. The arrangement is shown in Fig. 319, and consists of three gun-metal plates, each about $\frac{1}{16}$-inch thick; the two lower plates are perforated with holes, in circles *not* immediately over each other. It is usual to fit the discharge valves with a light brass spring, to return the valves to their seats after lifting. The lift allowed for moderate-sized valves is about $\frac{3}{8}$-inch. Other methods of valve fittings are numerous. One consists of a series of concentric corrugations on a metal plate, and is also frequently used. Within the limit of sufficient strength and durability, the lighter the suction valve can be made, the greater the possible efficiency of the air pump. The weight of the suction valve practically limits the vacuum attainable, for it is evident that, to be capable of lifting the valve, the pressure inside the condenser must be sufficiently in excess of that above the suction valve.

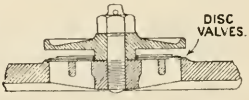


Fig. 319.
Kinghorn's Metallic Valves.

The efficiency of the air pump is a most important factor in the efficiency of the engine, and every opportunity should be taken to ensure its proper condition by examining and refitting the valves and packing. The pump should be so placed that, under ordinary working conditions, the suction valves are at a slightly lower level than the bottom of the condenser.

It is usual to make all parts of the air pump, its working barrel, plunger, valve seatings, and guards, of gun metal, except the plunger rod, which is made of rolled naval brass.

Air Vessel.—With a rapidly reciprocating pump which, like the air pump, is subjected to greatly varying load, the impact of discharge of an almost incompressible fluid, such as water, produces great shock on the discharge pipes and working parts of the pump. By using an air vessel, connected with the discharge, and in which some air is always imprisoned, the force of impact is reduced by the elastic cushion thus formed.

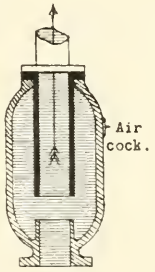


FIG. 320.
Air Vessel.

A type of air vessel is shown in Fig. 320. It is fitted on the air-pump discharge, and the more or less solid water has a clear passage through it. Once inside the upper part of the pipe, the air pressure contained in the vessel supports the weight of water above it, and serves to cushion the impact of the next discharge. The upper internal pipe is preferably made of zinc or cast iron, which, itself corroding, protects the gun-metal fitting and copper discharge pipe from corrosion. The air vessel should be fitted immediately above the discharge valves.

Edwards' Air Pump.—This pump differs from the ordinary single-acting bucket pump, as previously described, in the form and shape of the plunger, and the omission of both bucket valves and suction valves. The plunger is of the solid type, but is of hollow construction. The upper face is flat, and the lower part is made in the form of a cone, as shown in Fig. 321. The plunger works in a gun-metal barrel, which is surrounded by an exterior casing, in uninterrupted connection with the lower part of the condenser through the suction pipe. The casing at its lower part is similarly shaped to the plunger, and the plunger, when at the bottom of its stroke, nearly touches the casing.

The void left by the plunger on the up-stroke produces a flow of the condensed water and uncondensable vapours into the bottom of the casing ; and when the plunger descends it compresses the contents,

and forces them through holes in the working barrel into the space above the plunger. The casing must be suitably shaped to effect this purpose. The inducing action thus set up tends to continue the flow into the casing from the condenser. The plunger on the up-stroke catches the contents above it, closing the holes, and nearly the whole of the contents above this level are discharged during the up-stroke through the delivery valves. On the down-stroke the valves close, and an almost complete vacuum is formed above the plunger. As soon as the holes begin to open, the contents of the condenser flow into the barrel, and the action is continued by the forcing and inducing

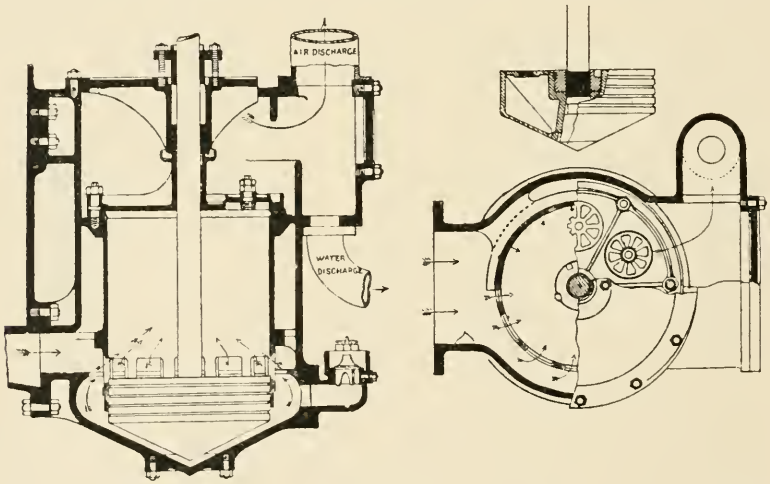


FIG. 321.—Edwards' Air Pump. (Two views.)

action of the plunger and the directing action of the casing. The flow from the condenser into the casing is almost continuous, and adds to the efficiency of the pump.

An air discharge pipe is fitted above the pump casing and water discharge pipe.

The plunger is not always fitted with packing rings, but water grooves are cut circumferentially in the working face, as shown. The rod is fixed to the plunger by a coned end, drawn up to its seating by a sunk nut on the upper side of the plunger ; and the nut is secured by set screws, partly fitted in the plunger and partly in the nut. The rod works through a long sleeve, which is water-packed for some part of its length, while the upper part is fitted with a stuffing-box and gland, cupped at its top end round the rod for moistening the packing.

The top end of the working barrel is fitted with a cover, in which the valves are seated. The cover is formed with a lip, so that the valves are always water-packed when the pump is working, and air leakage into the barrel is thus prevented. The valves are accessible when the pump is working, but, of course, there is some inconvenience and loss of water.

The lower part of the casing is fitted with a non-return relief valve, to prevent damage from any excess of pressure.

Edwards' air pump can be driven by any of the usual means, either from the main engines direct, or by independent engines. The efficiency obtained is generally greater than with the ordinary single-acting pump, and it is fitted in a large number of vessels of the mercantile marine, for both main and auxiliary purposes.

Independent Air Pumps.—Many vessels of large power are now fitted with air pumps worked by independent engines, and, following the same lines as in earlier days, when the feed pump was detached from the main engine, there appears to be some gain and advantage obtained by relieving the main engine of all duty except that of revolving the propeller shaft.

The advantages claimed are :

1. A vacuum can be obtained at all times, and its production allows the engines to be kept properly drained and ready for reversing or manœuvring.

2. The main engine is relieved from a large amount of work when starting.

3. The independent pump readily accommodates itself to any extra work thrown on it.

Dry Air Pumps. — For turbine installations an augmenting apparatus is commonly used in conjunction with the ordinary air pumps. For reasons which are more fully explained in Chapter XIV., a high vacuum is more advantageous with turbines than with reciprocating engines, and its attainment in some cases increases the efficiency of the steam by 5 to 8 per cent.

Messrs. Weir, D. B. Morison, and Dr. Weighton, particularly, have largely contributed to a much better efficiency of the condenser plant, and usually arrangements are now made for two suctions, one for the air and one for the heavier condensate which is mainly water. The condenser itself is now usually wedge-shaped, as shown in Fig. 9A and other illustrations. From these investigations the size and weight of condensers have been much reduced in recent years, because from

the greater efficiency of distribution of the tube cooling surface and other contingent improvements, the area necessary has in many instances been reduced to about one half of that usually fitted about fifteen years ago.

These new types of condenser require an improved air-pump system, and in many of them a type of air ejector is fitted in place of a dry air pump. Progress has been great since the first introduction of the Parsons vacuum augmentor in his early turbine installations, and in

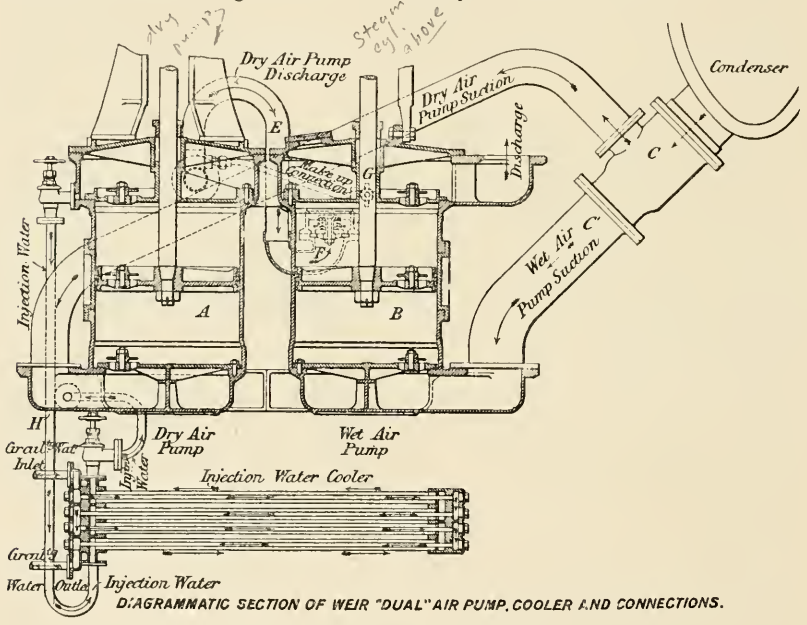


FIG. 322.—Weir's Dual Air Pump.

more recent plants the Weir Uniflow and the Contraflo systems in conjunction with a system for separately dealing with the dry and wet air condensate are commonly fitted. In the condenser itself there is also a great increase in the area of the eduction pipe or connection with the low pressure turbine, so that the rate of steam flow at this point is much reduced and it thus necessitates only a very small drop in pressure to maintain a steady flow between the two parts, the L.P. turbine and the condenser.

Weir Dual Air-Pump System.—Fig. 322 shows diagrammatically this system fitted in conjunction with a wedge-shaped Uniflow condenser in a general arrangement similar to that shown in Fig. 9A. The

two pumps, wet and dry, are actuated by a single steam cylinder immediately above the wet pump, *B*, and a system of rocking levers, worked off a crosshead, actuates the dry pump, *A*. One connection, *C*, is made with the bottom of the condenser, and a right-angle connection is made with *C* for the dry air pump. This pump discharges through the return pipe, *E*, and a spring-loaded valve, *F*, into the wet pump just below the head valves. *F* is adjusted to maintain 8 inches, of mercury, difference of pressure between the condenser and the discharge space of the dry pump. When starting the pump the filling valve, *G*, is opened for a short period to enable the vacuum to draw from the discharge space of the wet pump a supply of water for water-sealing, clearance filling, cooling, and vapour condensation in the dry pump. Then the valve, *G*, is closed, the water passes from the discharge space of the dry pump through the pipe, *H*, to an annular cooler, through which a supply of sea-water is circulated, and after being cooled the injection (fresh) water passes into the suction of the dry pump, which maintains a continuous circulation through the circuit, and any excess caused by condensation passes over to the wet pump. The advantages claimed for the dual system are, briefly: (1) No excessive cooling of the feed water. (2) The dry air pump discharges against only 4 lb. pressure instead of 15 lb. (atmospheric) pressure, as in the ordinary air-pump system. (3) The consequent reduction of load on the rocking levers economises the steam consumption for driving the air pumps. (4) The higher efficiency allows a smaller and lighter pump to be fitted.

Contraflo Condenser and Air-Pump System.—Fig. 323 shows this system as fitted in the s.s. *North Western Miller*, illustrated in Figs. 143 and 144, and applicable to other steamers whether reciprocating or turbine. The various parts making up this plant are shown in Figs. 324 to 326.

The steam exhausts from the L.P. cylinder into the main Contraflo condenser, *A*, where it is condensed and a vacuum produced by the mutual action of cooling and the air pump, *C*. At the lower part of the condenser is an air-pump temperature regulator, *B*, fitted in the main condensers when a reciprocating air pump is driven by the main engines. Its object is to increase, when required, the air-withdrawing capacity of the air pump and to prevent a fall of vacuum. This regulator is especially beneficial in hot climates, and it is also very useful in so regulating the temperature of the feed water that all the exhaust steam can be condensed by the feed water.

Under normal working conditions, as shown in the left-hand figure of Fig. 324, the aerated vapour passes through the devaporiser *A*, into the air pump suction pipe *B*. Condensate passes through the regulating valve *C*, into the pipe *B*. The air-withdrawing capacity of the pump is increased by closing or partially closing the valve and the temperature of the condensate lowered in the chamber *DD*, so that when the air leakage into the vacuum system is excessive, the air-withdrawing capacity of the air pump can be increased and the vacuum maintained. This arrangement is *not* fitted in turbine-driven vessels.

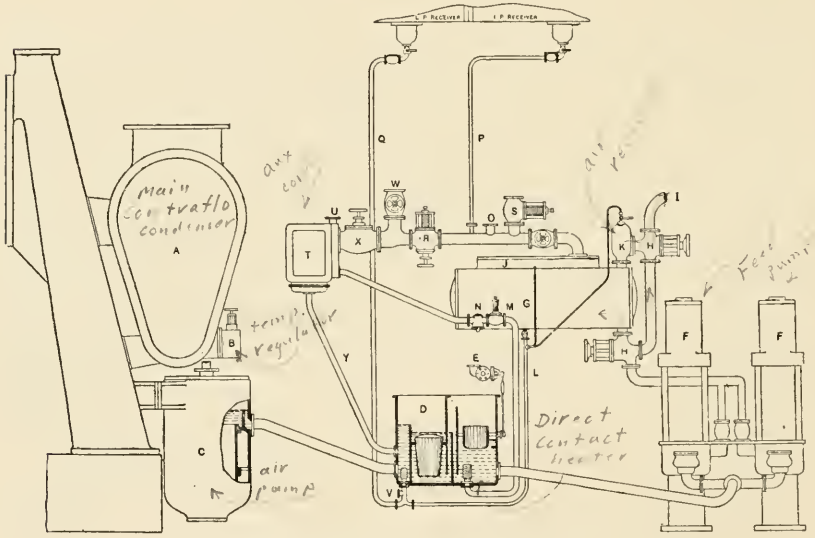


Fig. 323.—Diagram of Contraflo System.

The air pump *C*, in Fig. 323, discharges into the cascade filter, direct contact feed heater *D*, and float chamber, in which the float controls the steam supply valve *E* of the independent feed pumps *F*. (The detail of this arrangement is shown in Fig. 335.)

An exhaust steam compensator *surface* feed heater, *G* (Fig. 323), is fitted with valves *HH* on the discharge pipe of the pumps, one for direct discharge to the boilers through *I*, and the other for passage through the heater *G*, and thence through *K* and *I*. *K* is a receiver for collecting the air given off by the feed water before it reaches the heater, and is fitted with a drain pipe, *L*, which also serves for drainage from the heater *G*, thus keeping the heating tubes clean and efficient for long

voyages. *L* joins in with the drain pipe *Q* from the L.P. Receiver. and discharges into the same pocket as the air-pump discharge.

Exhaust steam is supplied to the surface heater *G*, from the M.P. Receiver drain pipe *P*, from the auxiliary engines and evaporator through *O*, and from the deck machinery, hoists, etc., through *W*. Any surplus steam can escape through a spring-loaded valve *R*, into the auxiliary contraflo condenser *T*, when the pressure in *G* is above

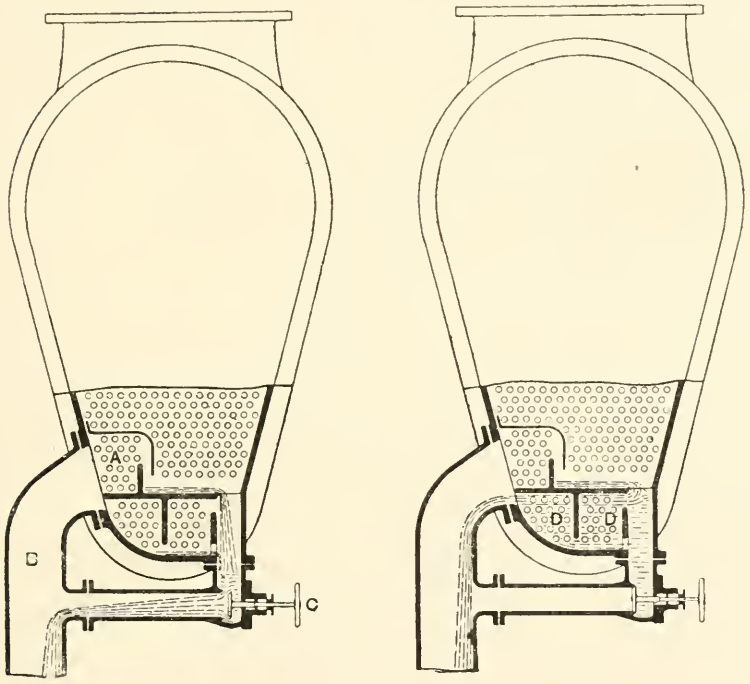


FIG. 324.—Contraflo Air Pump Temperature Regulator.

the working limit. In harbour, *R* is generally kept open to allow a free flow through *T*, and thence through a circulating cock, *N*, to the heater *G*. The drain for the heater *G* is led through a spring-loaded valve *M*, into the heating chamber of the cascade filter, and thus prevents any lodgment of air in the heater *G*. *U* is the exhaust pipe from the steering engine if on deck, but if the engine is in the main engine room, this exhaust is led direct into the heater *G*, with the other auxiliary exhausts. *S* is an escape valve to allow any excess pressure to escape to the waste pipe on the funnel.

The *contraflo auxiliary condenser*, of which an end view is shown at *T* in Fig. 323, is shown in section in Fig. 325, and has been fitted in many hundreds of merchant vessels. It is fitted to take any surplus exhaust not capable of being usefully employed in the compensator

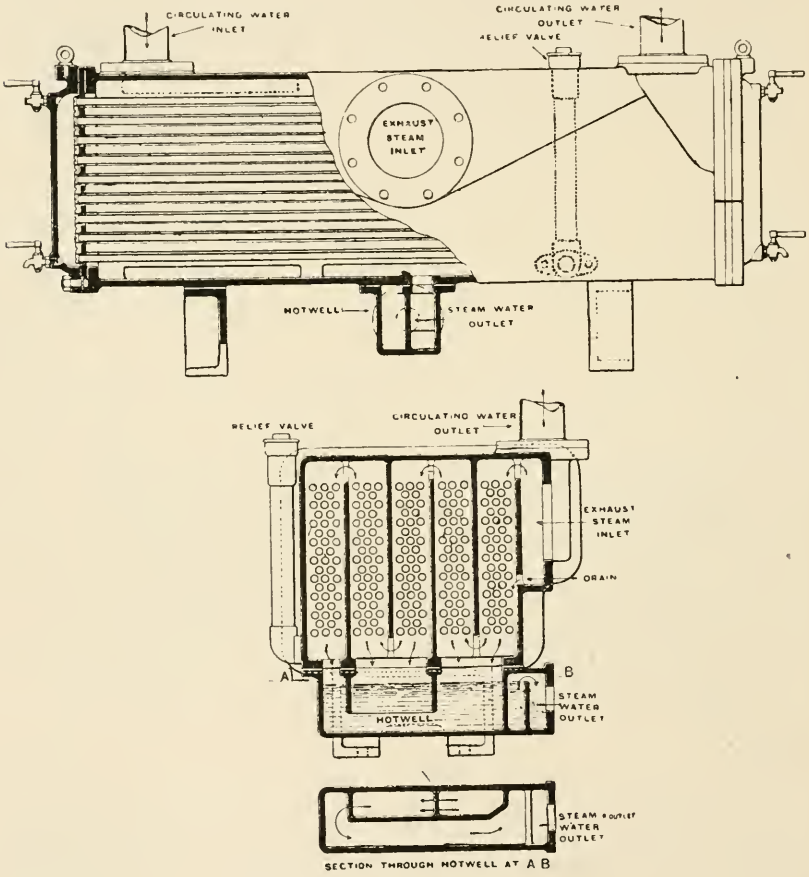


Fig. 325.—Contraflo Auxiliary Condenser.

heater, and its circulating water is supplied at sea from a branch of the main circulating discharge. The drainage from the auxiliary condenser is led into the cascade filter through *Y* (Fig. 323).

The whole contraflo arrangement, of utilising the exhaust steam and the drainage from the M.P. and L.P. Receivers for heating the

feed water, promotes considerable economy, as by its means the feed is raised to a temperature of over 200° F., or roughly it supplies about one-tenth of the total heat required for the conversion of the water taken from the condenser into steam for the next cycle of operations.

Another form of contraflo condenser, as fitted in geared turbine vessels by Messrs. Vickers, is shown in cross section in Fig. 314. It is placed immediately below the L.P. turbine and thus provides a very large exhaust inlet with the shortest possible eduction space. Initial condensation is obtained with a minimum penetration between the tube cooling surfaces. As the steam condenses and air presence becomes more pronounced, the cooling surface is divided into two or more parallel compartments by thin plates arranged and shaped so that they lie exactly in the line of steam flow through the condenser, and cause no resistance to its passage. These divisions also form wedge-shaped compartments, whose narrow ends meet at a position close to the condensate outlet and the devaporising chamber inlet. Thus air and non-condensable gases are generally concentrated towards the narrow outlet so that the air is of greatest density, and is consequently in the most desirable condition for withdrawal by the air pump. The kinetic energy of the inflowing steam materially assists in driving the air towards these narrow ends, and consequently the air region occupies a very small portion of the cooling surface. This localisation effects a very high transmission of heat through the cooling surfaces to the circulating water, and tends to produce a greater effect from a given area of surface for a given quantity of water circulated.

On its way to the air pump the air-vapour passes through a final compartment, known as the devaporising chamber, in which the vapour, still in association with the air, is further reduced by condensation, and the air cooled to a temperature approximating to the circulating water inlet, and thus the volume of the air is reduced to a minimum. A very high thermal efficiency is claimed for the high condensate or hot-well temperature obtained by this system, due to the enclosed steam being caught by the concentrating plates which prevent it from coming in contact with the colder layers of the condenser.

Contraflo Kinetic Reciprocating Air Pumps.—In connection with the condenser above described two pumps are necessary, one for the air and the other for the condensate ; this system is common in all turbine installations. In the contraflo system a steam ejector is used in conjunction with the dry air pump. Fig. 326 shows a general arrangement as fitted by Messrs. Vickers. There are two pumps

similarly actuated to the Weir dual air-pump system, and the steam ejector is fitted in connection with the left-hand pump. Without the ejector the pumps can be adapted for use as twin wet pumps, or as wet and dry pumps.

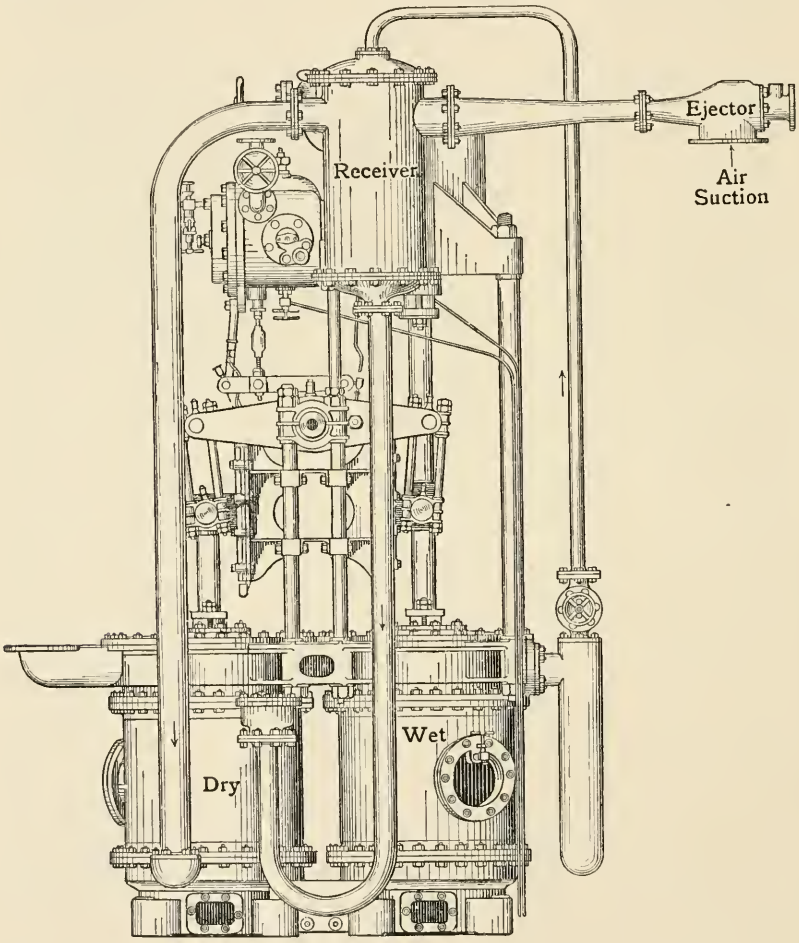


Fig. 326.—Contraflo Kinetic Reciprocating Air Pump. (Vickers Limited.)

The ejector withdraws the aerated vapour from the condenser, and, together with its own operating steam, discharges into the small receiver, where the vapour and steam are condensed by direct contact with a certain quantity of the main condensate taken from the dis-

charge of the water barrel, and discharged into the top of the receiver. The water in the receiver is drained away through a non-return valve on the air barrel (left). This valve is fitted below the head valves and above the bucket, so that water flows in on the down-stroke of the pump. The air which is withdrawn from the receiver by a separate connection with the bottom of the air barrel, and a small part of the condensate is supplied to this connection for the double purpose of providing sealing water for the pump valves, and for cooling the aerated vapour from the receiver to a temperature slightly above the main condenser temperature.

The vacuum created by the ejector in the condenser is generally 4 or 5 inches of mercury (say 2 to 2½ lb. per sq. inch) greater than that in the receiver which is maintained by the air pump, and in consequence the volume of air to be dealt with by the air barrel is enormously reduced as compared with the same weight of air in the condenser. Also its temperature is reduced, and in consequence its volume is further reduced, and the air barrel has an increased capacity for dealing with a given weight of air, which usually does not exceed 4 lb. per 10,000 lb. weight of steam condensed. This quantity may appear relatively small, but at condenser pressures it represents a very large volume.

The heat supplied by the ejector steam is absorbed by the feed water, and thus the water discharged by the air pump is in excess of the temperature of the water withdrawn from the condenser. As the pressure in the bottom of the air barrel is higher than the condenser pressure and the discharge pressure the same as the wet barrel, the pump is uniformly balanced from an operating point of view, and therefore runs smoothly and regularly at all speeds and vacuum conditions. Also the region of high vacuum is entirely apart from any mechanical movement, and consequently is not subject to the defects of wear and tear.

Leblanc Steam Operated Ejector Air Pump.—This ejector is made by the Mirlees-Watson Company and is illustrated in Fig. 327; it has been fitted successfully in the French destroyer *Boutefeu*, of which a general arrangement is shown in Fig. 328.

The ejector (Fig. 327) is arranged to work in two stages, *X* and *Y*. Steam is admitted at *C*, passes through a wire mesh strainer, and enters the steam chest supplying the six second-stage nozzles *Y*, and also through an auxiliary pipe to the first stage nozzles *X*. The air suction to the condenser is connected at *D*, and the steam and air discharged

at *E*. As a result of experiment several nozzles are fitted at each stage, and great care is taken by accurate machining to obtain the

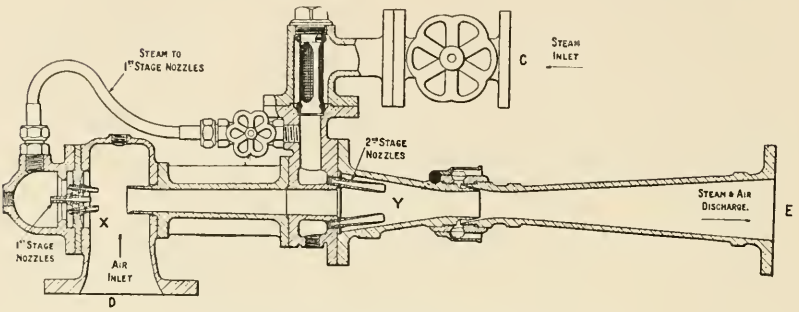


FIG. 327.—Le Blanc Steam Operated Ejector Air Pump.

correct angle of the face where the nozzles fit; otherwise the path of the issuing steam will not deal efficiently with the maximum quantity of air. A full description of the apparatus with calculations and

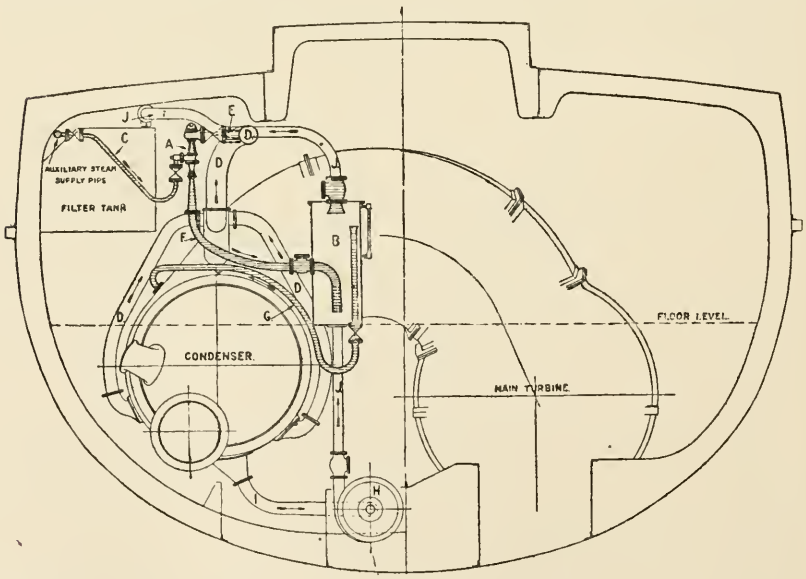


FIG. 328.—Le Blanc System fitted in Destroyer.

statistics and dimensions of the various parts, is given in a paper read by Mr. Ernest Jones before the Liverpool Engineering Society on

December 5, 1917. He states that the angle of taper of these multi-ejector steam nozzles is 6 degrees, and the angle of taper on the diffusers varies between 3 and 10 degrees as obtained by experiment. In a particular case he calculates the pump efficiency as 30·4 per cent, which compares favourably with any other types of pump working under similar conditions.

In the marine installation in the R.F. *Boutefeu* (Fig. 328), the ejector discharge *E* is carried to an intermediate tank *B*, placed above the discharge pipe *J* of the centrifugal condensate pump *H*. *B* is fitted with a return pipe *G* to the condenser, which is used when at anchor, and naturally the heat from the operating steam of the ejector cannot be utilised, and it is necessary to keep the condensate cooled by the condenser in a closed circuit.

The *Boutefeu* has two condensers, and each has connected with it a multi-ejector dry air pump and a centrifugal water extracting pump *H*. Cross connections are made between the two condensers, one to the air suction *D*, and another to the condensate suction *I*, so that at low speeds and powers a single set suffices for the two condensers.

It should be noticed that in the *Boutefeu* the two reciprocating pumps of other systems are not used, and that in consequence there is a considerable saving in weight and space with an apparently equal or greater economy of steam and heat, but this is very much dependent on the engine used for driving the centrifugal pump *H*.

Air and Steam Operated Ejector Air Pump.—In an arrangement fitted by Messrs. Brown-Boveri & Co., and somewhat similar to that shown in Fig. 327, atmospheric air is used instead of steam in the first stage (single) nozzle, and the second nozzle is operated by steam of reduced pressure. The system is shown in Fig. 329, and the discharge of air and steam is made to an auxiliary condenser. The extra weight of air admitted is inconsiderable in comparison with that withdrawn from the condenser by the ejector air pump, and does not interfere with efficient working.

The utilisation of atmospheric air as an operating agent for creating and increasing a vacuum in condensers is capable of still further development, but the principal objection is at present the separation of the steam vapour, which must be saved and condensed for use as feed water from the air, including that used as motive force. If the water vapour could be extracted the whole of the air could be discharged direct to the atmosphere or to the sea.

Turbo Driven Pumps.—In the *Boutefeu* the condensate pump

is of the centrifugal type, and a development of this system is desirable and has been fitted by the Brown-Boveri Company, in combination with a centrifugal circulating pump and a main oil circulating pump of the centrifugal type. A plan of this arrangement is shown in Fig. 330, fitted in connection with a geared-turbine set of marine machinery, and consists of one small turbine of the type described in Chapter XXXI., driving by means of pinions the three pumps mentioned above. The main circulating pump is driven at the low speed produced by the gearing, and the two other pumps at the turbine speed.

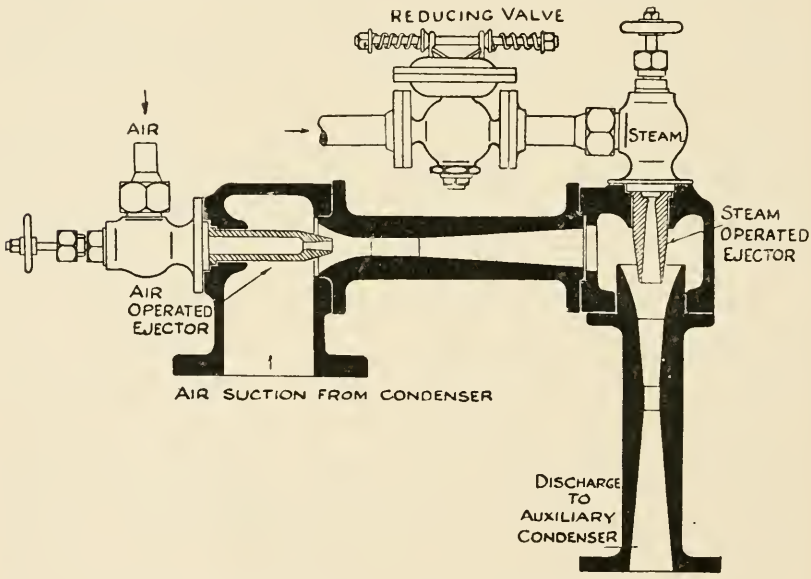


Fig. 329.—Brown-Boveri Air and Steam Operated Ejector Air Pump.

As in the case of all small turbines, the economy is good when the exhaust steam is utilised for feed heating, but in this case it would appear that the set as a whole would give about the same efficiency as would be obtained from three separate reciprocating engines, and would certainly occupy less space and be of less weight.

Circulating Pumps.—The pumps used for circulating the cooling water through the condensers are nearly always of the centrifugal type. The general arrangement of the inlet and outlet pipes is shown in Fig. 311, which also shows the position of the pump and the independent engine used to rotate it.

The *impeller*, or runner, which is shown in shaded section in Fig. 331, is fitted with a series of vanes, *A*, curved in such a way that the centrifugal action imparts a fairly high velocity to the water which is drawn in at *C* to the middle part *B* of the impeller, and thrown

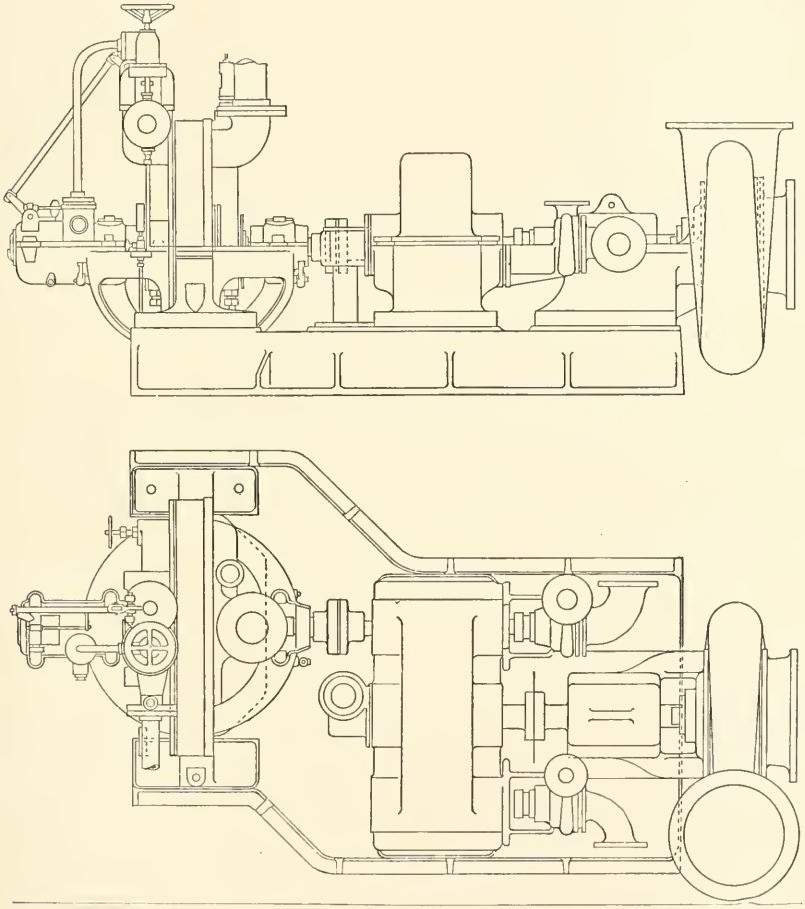


FIG. 330.—Turbine-driven Circulating, Condensate, and Oil Pumps. (Brown-Boveri & Co.)

off at the periphery or outer circumference *D*. The arrow heads show the direction of rotation and of the course of the water through the pump. There are no valves in the pump to get out of order; and as the pump is always situated some distance below the outer water-level, the pump chamber, suction, and inlet pipe are always kept

charged. The pressure the pump has to exert is very small, simply sufficient to overcome the internal friction of the pipes, tubes, and passages through which the water passes, and consequently a very large amount of water can be circulated through the condensers, with a high efficiency.

Each pump is generally driven by a separate engine, and a pair of pumps are usually fitted to each main engine. One pump can supply sufficient circulating water for all but the very highest powers, so that the circulating system is practically in duplicate. The shaft shown in the sketch runs in two lignum vitæ bearings. The bearing next the engine is also fitted with a stuffing-box and gland, to prevent

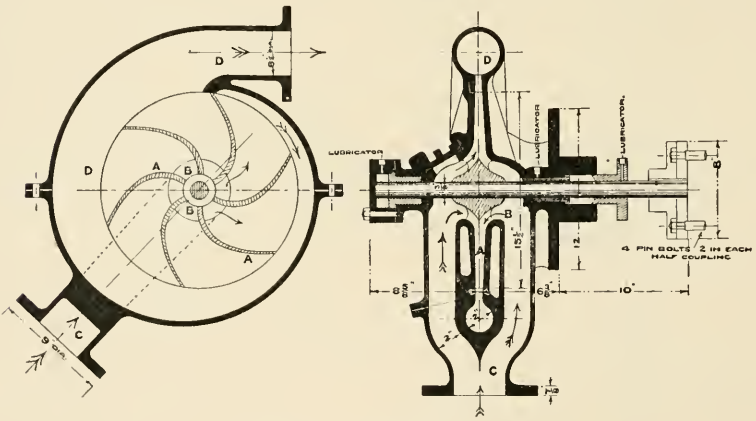


Fig. 331.—Centrifugal Pump.

the water leaking through ; the outer end of the shaft and its bearing are generally fitted with a cap, which encloses them. Both these bearings are cooled with a water service, and in the later types a pipe is led from the top of the pump chamber to the top of each bearing, to maintain a circulation through the lignum vitæ strips.

All parts of the pump and casing in contact with the sea-water are made of gun metal or bronze, except the lignum vitæ bearings already mentioned. A cock is fitted at the highest part of the pump chamber for letting out any air which may accumulate, and also for testing the efficiency of the pump by observing the height to which the water is thrown. Any decrease in the usual height generally points to some choking of either the inlet valve or pipe.

In Fig. 332 a somewhat different arrangement of the circulating-

pump bearings is shown. The inlet water enters from both sides of the impeller, and the pumps are generally made in this way. The impeller shaft is fitted with outside bearings of gun metal, with white-metal rubbing surfaces, which are lubricated in the usual way. Stuffing-boxes and glands are fitted to keep the shaft water-tight where it passes through the pump casing. This arrangement is particularly useful when navigating in shallow waters, such as the Suez Canal, as the sand and grit carried in with the circulating water is thus excluded from the bearings; it is adopted in the *Mauretania*. In this type of pump it will be noticed that the ends of the impelling blades terminate at the inner part of the outlet tube, and for the same diameter of casing a smaller diameter of impeller is used.

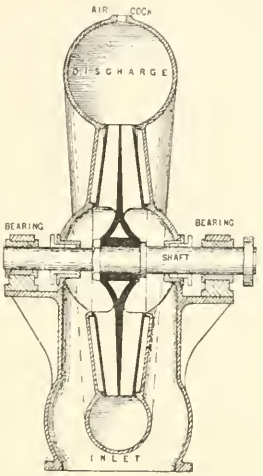


FIG. 332.—Centrifugal Pump.

Circulating Bilge Suction.—To each pump a connection is fitted to draw water, if required, from the engine-room bilge. If required for pumping out the stokeholds, these can be drained by isolated independent pipes to either engine-room bilge, and then be pumped out from there. The arrangement is shown in Fig. 311, which also shows the rods for the working of the valves. The handles or wheels should be accessible when the engine room is flooded, while still leaving the centrifugal pumping engine free to be worked. Unless particular attention is paid to the design of the suction valve so that it can lift easily and accommodate itself to the varying fluctuations of the pump and the run of water to the suction, no great reliance can be placed on the centrifugal circulating pumps to remove much water from the bilge. Apart from the inefficiency of the pump itself as a suction pump, there are numerous obstacles in the engine room, such as frames and pipes, which all prevent a good flow to the suction, and render its efficacy doubtful. In recently constructed vessels, larger holes in the framing are made to allow a better flow to the suctions.

A large ship is generally fitted with four or more circulating pumps, and their united capacity is from about 20,000 to 40,000 tons per hour. This apparently great quantity is very small compared with the amount

of water which will enter a hole of very moderate size at a depth of about 20 to 25 feet below the water-line (see Chapter XXX.). In emergency every effort should be made to reduce the flow of water into the ship, and then a possibility exists for keeping the reduced leakage within safe limits. The pumps are generally practically tested in the ship with a reduced steam pressure and number of revolutions.

Heat rejected in the Condenser.—The exhaust steam enters the condenser at about 1 to 2 lb. absolute pressure per square inch, corresponding to about a temperature of 120°. This steam in practice contains by weight at least 25 per cent of moisture in suspension. If considered as dry saturated steam, then every 1 lb. weight of steam carries with it into the condenser about 1030 b.t. units in the form of latent heat, and, in addition, the cooling water reduces the temperature to about 90°, the temperature of the feed water. Therefore the heat carried away by the circulating water, or the heat rejected in the condenser, is for each 1 lb. of steam used about—

$$1030 + 30 = 1060 \text{ b.t. units.}$$

For a modern battleship or cruiser, about 10 lb. weight of steam is used per S.H.P. for the main engines, and about 2 lb. for the auxiliary engines, per hour. Taking the amount at 12 lb., the heat rejected is (considering the exhaust to be dry saturated)—

$$1060 \times 12 = 12{,}720 \text{ b.t. units per S.H.P. per hour.}$$

Quantity of Cooling Water required.—The heat rejected in the condenser decides the quantity of water required to cool the exhaust steam and condense it into water. A rise of about 25° in temperature is allowed for the circulating water, and assuming that each 1 lb. weight of steam conveys 1060 units of heat into the condenser, the quantity of cooling water required is—

$$\frac{1060}{25} = 42{\cdot}4 \text{ lb. per 1 lb. weight of steam used by the engine.}$$

If n lb. of steam are used per I.H.P. per hour, then the quantity of cooling water is—

$$= n \times 42{\cdot}4 \times \text{I.H.P. per hour (weight in lb.),}$$

and the circulating pumps must be capable of forcing about this amount through the condensers per hour.

In the Tropics and narrow waters the temperature of the sea is

from 80° to 85°, and if a lower condenser temperature is required, it is obvious that the quantity above must be increased. With a temperature of discharge of about 105°, a vacuum of about 28 inches is possible ; and in practice the quantity stated above is considered more than sufficient, and about 40 lb. is usually allowed for purposes of design of piston engines ; but for turbines a much greater quantity is considered necessary.

CHAPTER XXV

THE FEED WATER AND FEED PUMPS

Sources of Supply.—The water condensed in the condenser is pumped out of it by the air pump, which delivers it into either the hot-well tank, from which it is pumped by the hot-well pump into the feed filter, or into the filter (direct as shown in Figs. 333 and 334); from the filter it flows into the feed tank. From the feed tank the boiler feed pumps take their supply and return the water to the boiler. The feed tanks are also supplied from the auxiliary air pumps, and from the drain tank by its pump, but usually both these supplies are passed through the filter on their way to the feed tank.

The blow-down, which is generally automatic, and fitted on the steam separator in the steam pipe just previous to its connection with the engine, is also a source of supply to the condenser as a water-saving appliance. Any surplus steam can be passed through it, as a silent blow-off, and the water, instead of being lost up the waste-steam pipe, is saved by passing it through the condenser.

The water-saving appliances are thus—

1. Main and auxiliary condensers.
2. Silent blow-off, or automatic drain to main steam pipe.
3. Steam drain system; the water is collected in a drain tank from steam traps fitted at various parts, and pumped through the filter to the feed tank.

The whole of the water which leaves the boiler as steam cannot all be saved; there must be some loss from leakage, and this is made up from the reserve feed tanks or from the evaporators.

Necessity of Fresh-water Feed.—Sea-water contains about $\frac{1}{32}$ part by weight of solid matter, in which common salt predominates but is not generally deposited on the heating surfaces of the boiler, and is easily removed by blowing down.

Sea-water contains about, in lb. per ton (2240 lb.)—

58	of common salt (chloride of sodium).
5	,, sulphate of magnesia.
8	,, chloride of magnesia.
$\frac{1}{4}$	,, carbonate of lime and magnesia.
$3\frac{1}{2}$	,, sulphate of lime.
$1\frac{1}{4}$	,, organic matter.
Total 76	lb.

Of these, the chloride of magnesia is the most dangerous. At a temperature of 212° it decomposes into magnesia and hydrochloric acid. The latter is an active corrosive agent ; but the corrosive effect is neutralised by adding alkali, usually lime, to the feed water, which should always be alkaline.

Sulphate of lime (calcic sulphate) is the most troublesome deposit, with the exception of oil. At a temperature of about 280°, corresponding to a pressure of about 40 lb., sea-water is incapable of holding sulphate of lime in solution.

Carbonate of lime forms, with the sulphate of lime, about 85 per cent of the scale usually deposited in the boiler. The carbonate, although it is not freely deposited until a temperature of 350° is reached, corresponding to a pressure of 120 lb., is, however, deposited at a lower density than the sulphate, and is generally first deposited on the heating surfaces. It is followed by sulphate of lime, salts of magnesia, silica, and common salt, in the order named ; but the two last are very seldom found, unless the density has been very high, and above that usually maintained, even when the feed water is nearly of the consistency of sea-water.

About 1 lb. per ton weight of *sea-water*, density 10, fed into the boilers is left as scale deposited on the heating surfaces, when the boilers are blown down at regular intervals according to the regular practice.

The rate of conduction of heat through a clean lime scale is only $2\frac{1}{2}$ per cent of that of steel, such as is used in a furnace. By practical experiment it has been shown that with $\frac{1}{16}$-inch of scale 15 per cent more fuel is required, and with $\frac{1}{4}$-inch of scale about 60 per cent. With a perfectly clean furnace of ordinary thickness, the temperature of the water surface of the plate may be only 15° in excess of the steam generated ; but with a scale of $\frac{1}{2}$-inch thickness the temperature of

the surface must be raised to nearly double that of the steam generated. With an oil scale, the temperatures are exceedingly high in the metal of the furnace compared with that of the steam generated. It is an oil scale which is the greatest source of danger in boilers, and for continuous working a grease filter is necessary.

At temperatures above 600°, steel tends to become carbonised into cast iron with a consequent reduction of tensile strength.

It is for these reasons (the prevention of scale forming and consequent overheating of the boiler plating, and for promotion of economy) that sea-water is excluded as much as possible from high-pressure boilers.

Quantity of Water pumped.—The weight of steam used must be returned to the boiler as feed water to maintain the same safe working level. This amount may be taken as 10 to 16 lb. per S.H.P. per hour. For example, take the S.H.P. as 11,200, and the weight used per S.H.P. per hour as 12 lb. Then in every 24 hours the total feed water pumped is—

$$11{,}200 \times 12 \times 24 \div 2240 = 1440 \text{ tons};$$

or, quantity of water pumped is 130 tons per 1000 S.H.P. per 24 hours.

Feed-water Loss.—Under good conditions of working, and with well-appointed saving appliances, the loss should not exceed about 3 tons per 1000 S.H.P. per 24 hours, so that the actual loss of water pumped is only about 2 per cent. Unfortunately, the loss is frequently greater, and, with high pressures, the tendency to leakage is also greater. Ships vary considerably in the arrangement of machinery and steam pipes, and the number of auxiliary engines is very great, particularly in warships. This method of computing the loss is very unreliable, but it is the nearest yet suggested, and with improved simplicity of arrangement the error becomes less.

Feed Tank and Fittings.—The feed water gravitates from the filter, or it is pumped, in the case of a pressure filter, into the feed tank. One feed tank is fitted to each main engine, and they are connected with each other through a large pipe with a shut-off valve at each tank.

The general arrangement of the feed tank and its connections is shown in Fig. 333. They are described below, with their uses :

(*a*) Gauge glass, extending from nearly the top to the bottom of the tank, to indicate the quantity of water in it.

(*b*) Index plate, fitted near the gauge, to show the quantity in *tons* of water in the tank.

(*c*) Connection with tank in other engine room, with a shut-off valve next each tank. In some earlier vessels only one valve (a sluice) is fitted.

(*d*) Inflow pipe from the grease extractor, or filter. Means are sometimes fitted by which the water can be diverted into either tank.

(*e*) Filling pipe from alongside, led into the top of the tank, and fitted at some point above its upper part with a steam injector and

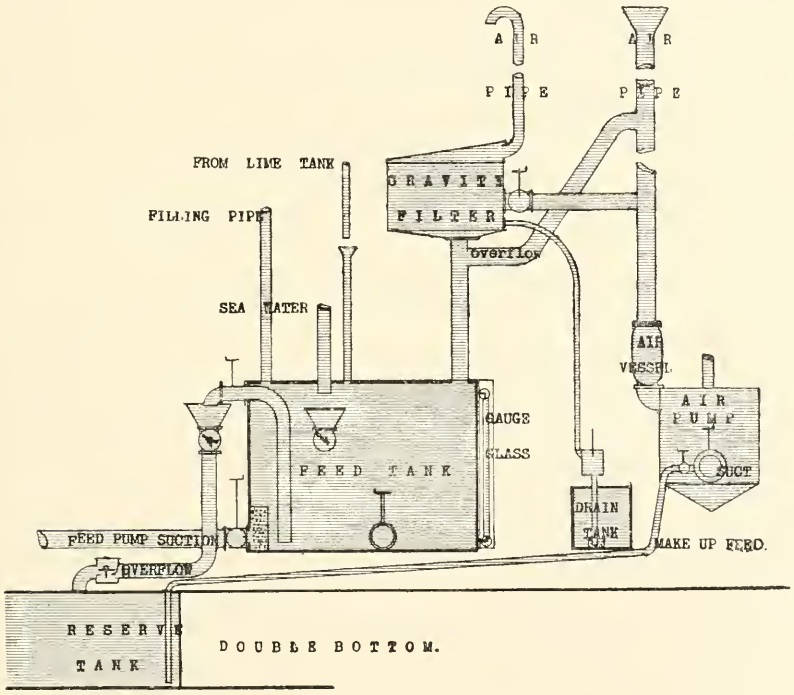


FIG. 333.—Feed Tank and Connections.

pipe for drawing the water up from a tank or vessel alongside, and discharging it into this pipe through a bell-mouthed funnel, and so to the tank.

(*f*) Supply from the sea, led into a funnel-shaped vessel on the side of the tank, and fitted with a shut-off valve. The water is obtained either from the fire main discharge by connecting on a hose, or by a pipe fixed in a convenient position for the purpose and which can be diverted from dripping into the funnel when it is not required. The

top of the funnel is kept below the overflow from the feed to the reserve tank, so that sea-water cannot enter the latter.

(*g*) Supply from the lime tank, led into the top of the tank, preferably through a funnel, so that the passing water can be seen, as with the alongside and sea-water supplies. The lime is precipitated in a small tank in the upper part of the engine room, and the clean lime water gravitates from the upper part of this tank into the feed tank, leaving behind as much as possible of the sediment.

(*h*) Overflow to reserve tanks, fitted with an internal pipe leading to the bottom of the feed tank so as to allow as far as possible only clean water to pass through the overflow. A shut-off valve is fitted at the outlet, which is led into a funnel and pipe leading to the reserve tanks; a shut-off valve is fitted just below the funnel, and a non-return valve is fitted next the reserve tank. Arrangements are made so that any one of the reserve tanks can be filled by opening or closing the proper valves.

(*i*) Feed-pump suction pipe, fitted to connect the bottom of the tank with the feed pump, with a necessary cross connection for pumping from either tank with any pump. A shut-off valve is fitted next the tank, and a strainer inside the tank over the pipe orifice. The position of the feed tank should be such that there is a free flow through the pipe into the pumps; the pump suction boxes are therefore placed as low as conveniently possible, and the feed tank about 3 feet above the suction boxes of the pumps.

The sides, ends, and top and bottom of the tank are flat surfaces, and it is generally necessary to fit stays horizontally from side to side, but only occasionally from end to end and from top to bottom. A manhole door, fitted internally, and similar to those used for boilers, is also fitted. For protecting the plating from galvanic action, slabs of zinc are connected electrically with the stays or other convenient parts. A drain valve is fitted for use when required to totally empty the tank. If possible, it should be in sight from the starting platform, so that any leakage may be seen.

Reserve Feed Tanks.—In nearly all cases where the ship is built with a double bottom, some of the spaces thus created are utilised as tanks for the storage of fresh water for use as boiler feed. The spaces used are generally situated about the mid-length of the vessel; in cargo steamers the forward and after ballast tanks are also used, as convenient.

The arrangements for filling are as already described, *i.e.* through

the feed tanks from alongside ; the gained water from the evaporators passes through the condenser, feed tank, and overflow, with the other condensed water. Open-ended air outlet pipes are led from each tank at about their highest points. The auxiliary feed pumps can draw water from any one tank by opening suitable valves, and also from the feed tanks ; but the main feed pumps usually pump from the feed tanks only. A safety valve, loaded to about 50 lb. per square inch, is fitted to each pump suction to prevent overloading from any back pressure, and so bursting the pipe.

Zinc protectors are suspended in each tank, as in the feed tanks, fitted in electrical contact with the ship plating. The reserve and feed tanks are not painted internally, but kept clean and free from accumulations of oxide scale ; and after cleaning, they are rubbed down

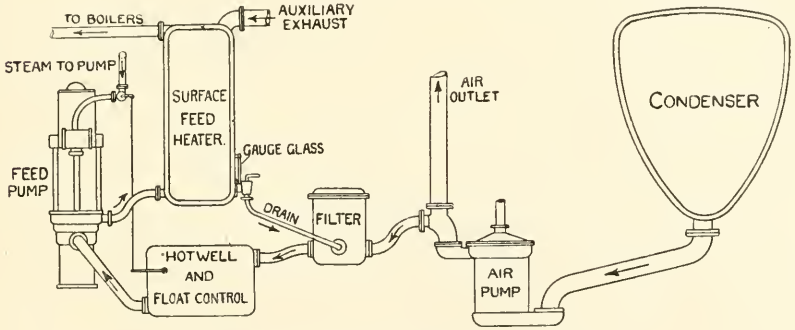


FIG. 334.—Diagram of Surface Feed-heating System.

with mineral oil and dried off as much as possible with a wad. Care must be taken to prevent any loose waste or oakum remaining to choke the feed suctions.

Feed-water Systems.—Fig. 334 shows an arrangement similar to that of Fig. 333, but connected with a Weir wedge-shaped condenser, filter, hot-well, or feed tank, and a surface feed-water heater of the pressure type fitted between the feed pump delivery and the boiler.

The pressure type of surface heater has some mechanical disadvantages as compared with the direct contact system, but the feed-water can be heated to a higher temperature corresponding to, or slightly less than, the temperature of evaporation or boiling-point in the boiler.

For example, with a boiler working pressure of 210 lb. and a saturated steam temperature of 391° F., and a condensate temperature of

120°, the latent heat of the steam is 841 b.t. units, and the sensible heat, 391 – 120, or 271 b.t.u. per 1 lb. of water or steam, and the ratio is :

$$\frac{271}{841+271}=\frac{271}{1112}=24{\cdot}37 \text{ per cent},$$

which may be considered the absolute maximum of feed heating attainable in practice, and the actual economy must be much below this because there must be some losses from radiation, etc., and live steam taken direct from the boiler must be used in whole or in part

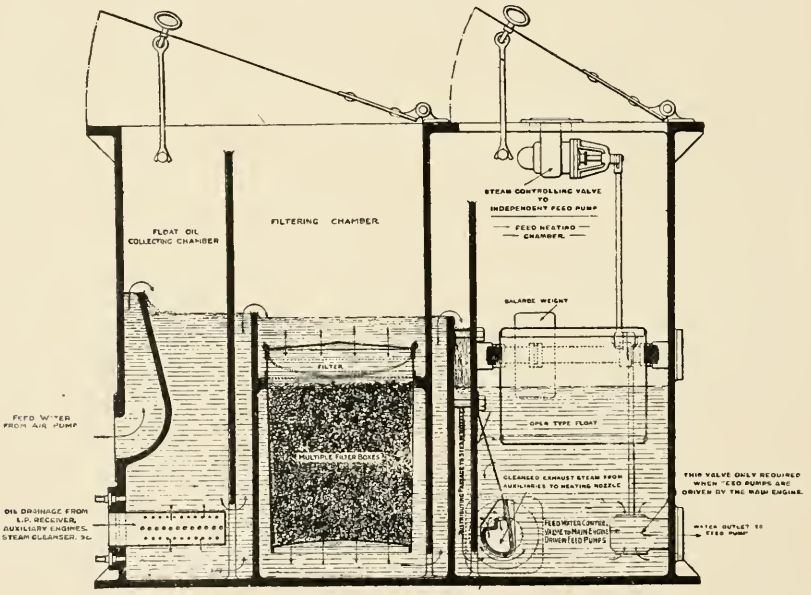

FLOAT OIL COLLECTING CHAMBER
FILTERING CHAMBER
STEAM CONTROLLING VALVE TO INDEPENDENT FEED PUMP
FEED HEATING CHAMBER
BALANCE WEIGHT
FILTER
MULTIPLE FILTER BOXES
OPEN TYPE FLOAT
FEED WATER FROM AIR PUMP
OIL DRAINAGE FROM L.P. RECEIVER, AUXILIARY ENGINES, STEAM CLEANSER, &c.
CLEANED EXHAUST STEAM FROM OVERFLOWS TO HEATING NOZZLE
THIS VALVE ONLY REQUIRED WHEN FEED PUMPS ARE DRIVEN BY THE MAIN ENGINE.
WATER OUTLET TO FEED PUMP

Fig. 335.—Cascade Filter.

to attain such a high temperature. Generally exhaust steam is used, and the feed-water may then be heated up to about 210° as a maximum, which gives a ratio of

$$\frac{90}{841+90}=\frac{90}{931}=9{\cdot}5 \text{ per cent},$$

which may be considered the maximum efficiency under the conditions stated. From this percentage about 0·5 per cent must be deducted for the extra work thrown on the feed pumps to push the water through the surface heater.

Fig. 335 shows the detail of the cascade filter, heater and control

tank, as fitted in the s.s. *North Western Miller*. The apparatus is divided into three compartments: the first receives the air-pump discharge and the drainage from all sources; the second contains portable filter buckets; and the third is fitted with a heating nozzle in series connection with the compensator heater (surface type), Fig. 323, and also a float which controls the feed pumps.

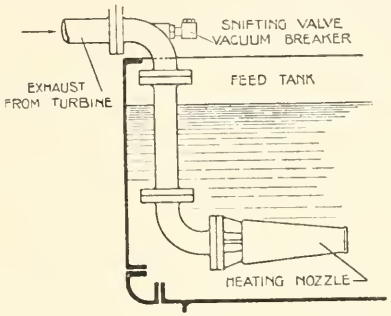


FIG. 336.
Weir Exhaust Steam Contact Feed Heater.

Fig. 336 shows in detail a Weir direct contact exhaust steam feed heater for utilising the exhaust steam coming from a turbine of the type described in connection with Figs. 344 and 345. Exhaust steam coming from a turbine is free from oil and therefore it can be used direct without filtering; a filter would be necessary for reciprocating engines to clear the oil from the water. The object of the nozzle shown inside the tank is to promote circulation in the tank with convection heating. The snifting valve prevents the return of water to the turbine through the exhaust.

Hot-Well and Pump.—In consequence of the increased work thrown on the air pumps when they were also required to deliver the condensed water through the filter into the feed tank, it was considered advisable to fit a pump for this special purpose, and so relieve the air pump of the unnecessary stress. The air pump therefore delivered its contents into the hot-well, which was fitted in close proximity, and the hot-well pump then forced the water up to the filter (with a pressure filter it also forces it through the filtering medium), from which it flowed into the feed tank.

The hot-well, if fitted, is of small capacity, and an automatic arrangement is generally fitted to the pump, so that it is governed by the height of water in the tank. The steam-cock of the engine driving the pump is attached to a system of levers, which are operated by a float inside the tank; any increase of water in the tank opens more steam to the engine, and any decrease has an opposite effect. An air outlet pipe and an overflow pipe are both fitted to the tank, so that, in the event of its becoming too full, the water can escape to the feed tank or reserve tank. Generally one hot-well and pump were

fitted in each engine room, when arranged for their use. The adoption of a gravity filter, a better type of air vessel on the air pump, and finally the adoption of an independent air pump, have each and all contributed to the abolition of the hot-well and pump.

Feed-water Filter or Grease Extractor.—The filter takes its supply from three sources :

1. Main air pump, or hot-well pump, if fitted.
2. Auxiliary air pump, if fitted ; with the independent air pump it is unnecessary.
3. Drain-tank pump.

There are several varieties of filters. Under favourable conditions

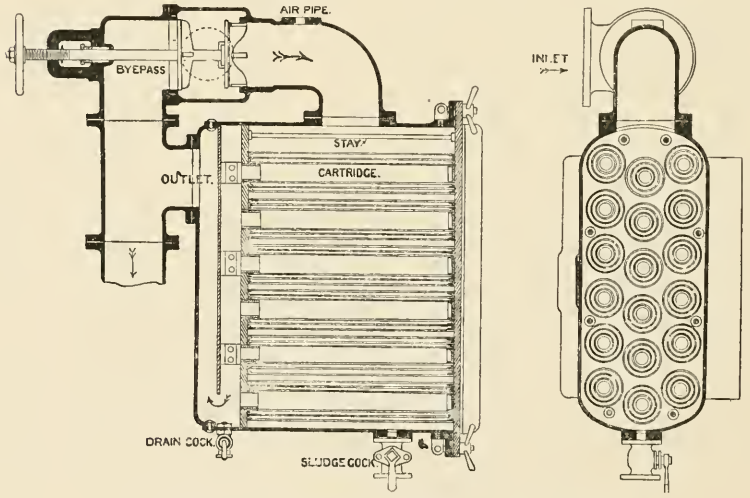


Fig. 337.—Rankine Feed-water Filter or Grease Extractor.

there is no necessity for a hot-well and pump, as the filter itself takes the place of the former, which simply acts as a quiet place in which the air can be evolved from the water and escape to the outer air or atmosphere.

A *gravity filter* is generally placed as near as convenient to the air pump and at a fair height above the feed tank, so as to give a steady gravitation of the water into the latter and an uninterrupted and non-devious flow through the air-pump discharge.

Inside the Rankine filter (Fig. 337) there are several cells, on frames, in each of which three separate pieces of filtering material, generally of coarse towelling, are fitted. The cells are of circular shape. A water-tight cover is fitted on the end of the filter, and is

fastened in its place by hinged screws and butterfly nuts. After removing the cover, the cartridges, or any separate stratum, can be removed, and new filtering material can be fitted. A spare set of cartridges is supplied, and can be put in at once; this operation should only take a few minutes if everything is previously prepared. An air pipe is led away from the highest available part of the filter to a height of about 4 feet. About $1\frac{1}{2}$ to 2 square inches of total surface of the three strata through which the water passes are allowed per I.H.P. A bye-pass valve is fitted for use when opening the filter.

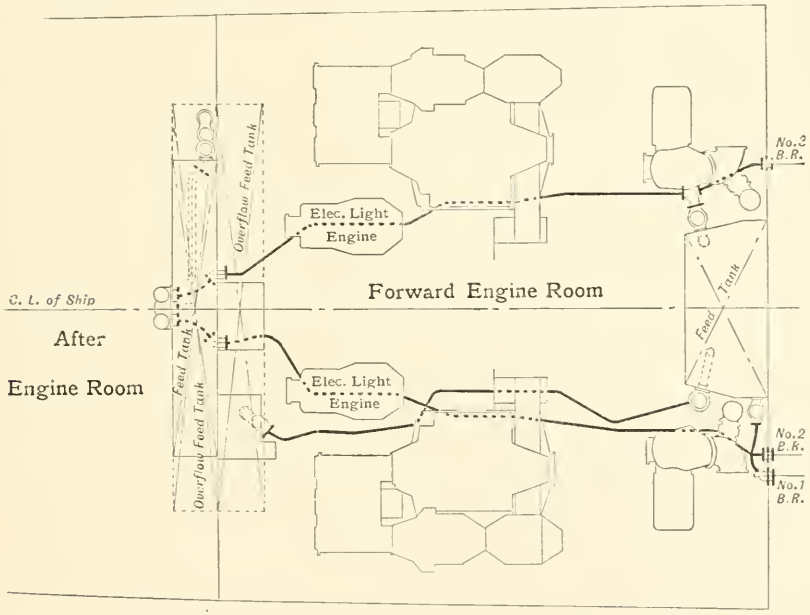


FIG. 338.—Feed Suction Pipes in Engine Room. (Beardmore & Co.)

As shown in Fig. 333, the main inflow pipe to the filter is fitted with a shut-off valve, and when this is shut the water is pumped to a higher level and passes through a pipe connected directly with the outflow pipe to the feed tank. This pipe prevents any great influx of water overloading the air pump or filter.

Feed Suctions and Pipes.—The principle already described is adopted for modern naval vessels, but the positions and general arrangements differ in appearance according to the internal fittings. Fig. 338 shows the general arrangement in a cruiser's engine rooms, and Fig. 339 that in the boiler rooms as recently fitted.

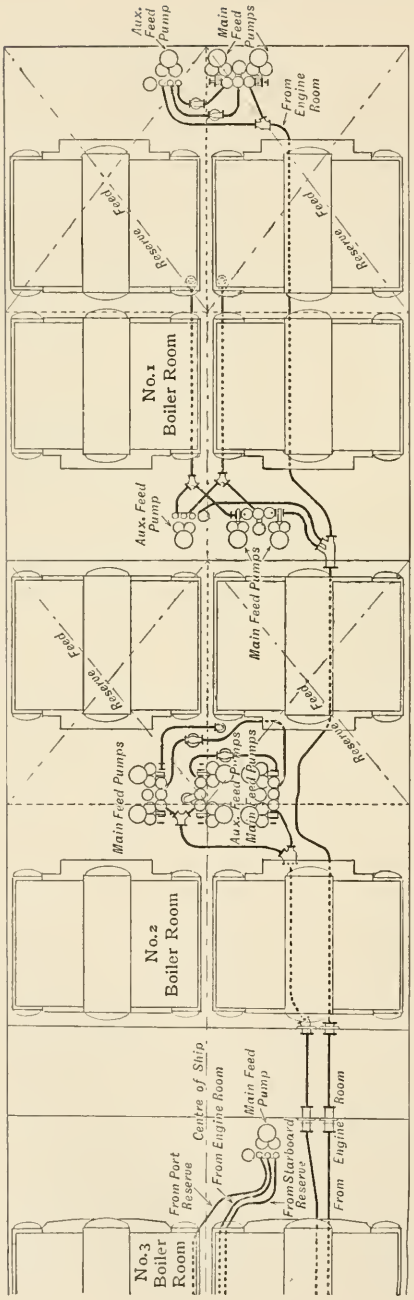


Fig. 339.—Feed Suction System, Water Tube Boilers. (Beardmore & Co.)

All the feed delivery pipes are fitted in the boiler rooms, and therefore comparatively short and simple, but the suction pipes which are not under pressure, except accidentally, are long, and extend the whole length of the respective boiler and engine rooms. A feed tank is fitted at the forward end of each engine room, and there are two overflow tanks. The forward, or No. 1, boiler room suction is connected directly with the forward feed tank. No. 2 boiler room is connected directly with the after-feed tank, and No. 3 boiler room is connected with both forward and after tanks; and the part of its pipe between the two tanks forms a workable connection for the other boiler rooms when required.

In No. 1 boiler room there are four main feed pumps and two auxiliary feed pumps equally divided between the two stokeholds, the boilers being arranged back to back. In No. 2 boiler room the boilers are face to face, and thus there is only one common stokehold containing four main feed and two auxiliary feed pumps. The arrangement in No. 3

boiler room is similar to No. 1, but only part of this boiler room is shown in the illustration.

Parts of the double bottom spaces under each boiler room are set apart as port and starboard reserve feed tanks, thus considerably increasing the supply of fresh feed water, in addition to the evaporators, which can be used to *make up* the loss under steam.

Each feed pump, main and auxiliary, has three suctions controlled by separate valves: port reserve tank, main feed tank, and starboard reserve tank. The arrangement is thus simple and practically universal.

The position of the pumps in Nos. 1 and 2 boiler rooms is chosen to suit oil fuel burning only. In No. 3 boiler room space is allowed for both coal and oil fuel firing.

The feed delivery pipes are all in sight from the stokehold platforms, and the usual arrangement of the main feed entering the boiler on the right and the auxiliary feed entering on the left of the boiler in each case (in this instance the feed enters the steam drum of the Yarrow type boilers) is generally carried out.

Feed Pumps.—The feed pumps are a necessary adjunct of the main engines, as it is by their means that the water is returned to the boilers after condensation from the steam supplied to the engine; it thus completes the cycle of operations.

1. The earliest types of marine engines were generally fitted with a lever attached to, and worked by, the end of the piston rod; this lever worked the main feed pump, and in consequence the pump only worked at the same time as the engine. For supplying the boilers when the main engines were not at work, a donkey pump was generally fitted in the stokehold, and worked by an independent engine. This system of working the feed pump from the main engine is still used in steamboats and other small installations.

2. The increasing rate of revolution of the main engine led to the almost general adoption of independent feed engines for both main and auxiliary use. The main feed pumps were usually situated in the engine room, and the rate of supply was regulated automatically. A cock was fitted in the steam pipe to the feed engine, and connected by a simple system of levers with a float in the feed tank; a rise of water-level in the tank opened the cock and put the pump on faster, and a fall correspondingly eased it down. This arrangement was most satisfactory, and generally adopted for all vessels with cylindrical boilers, and is still common in the mercantile marine.

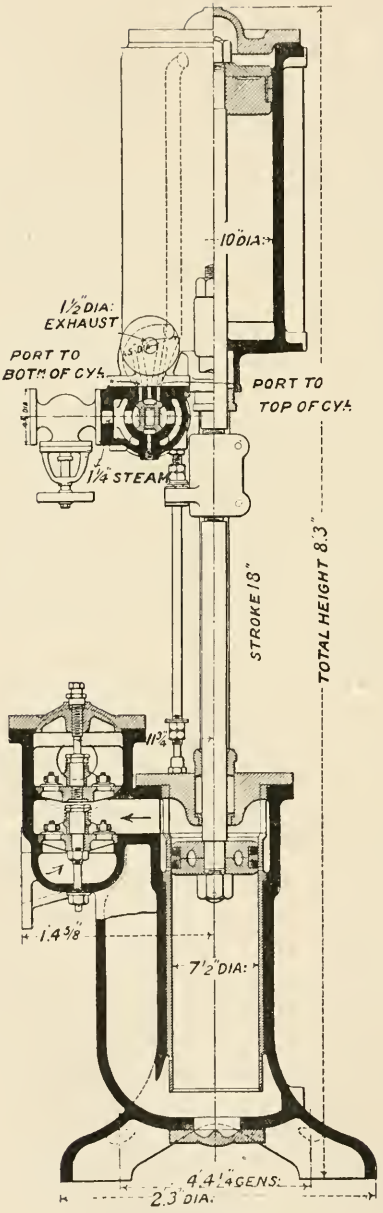


Fig. 340.—Bradford Feed Pump.

3. The rise of pressure, obtained at first by the use of water-tube boilers, and the increased total power, led to the removal of the main feed pumps to the boiler rooms. The work done by the pumps is thus decreased with the shorter delivery pipes and consequent less frictional resistance; and the control of the feed water is under the supervision of the boiler-room staff, in its own compartment.

The independent pumps are generally of the single-cylinder variety of direct-acting pump (Fig. 341); but the centrifugal type driven by a small turbine is becoming increasingly used.

Bradford Feed Pump.—The Bradford feed engine is distinctive in that there is neither a shuttle valve nor an auxiliary valve. The slide valve is given an intermittent rocking motion, derived from the crosshead when near the ends of its stroke. As shown in Fig. 340, the piston has nearly completed its upward stroke and is rotating the valve with an anti-clockwise motion, so as to connect the bottom of the cylinder with the central or exhaust port, and at the same time to open the right-hand port to supply the top of the cylinder with steam.

The pump-barrel is lined with gun metal, and the bucket is of the same material and fitted with packing. The essential difference

between the action of this pump and others of a general similarity of type is that holes are drilled in the liner-barrel so that they are just overrun by the bucket, and they thus relieve the delivery pressure in the clearance space for a moment at the reversal of the motion of the bucket. This arrangement allows time for the delivery valves to seat themselves without shock. The pump valves are usually of the flat gun-metal disc type, which is not generally suitable for delivery pressures of 180 lb. per square inch, but of course valves of the Weir solid type can be provided in the case of higher pressures.

Weir Monotype Feed Pump.—Fig. 341 shows a sectional elevation of one of these pumps as fitted in large vessels. The pump is direct-acting, with a single steam cylinder and a double-acting water end. The valve action is positive, and the pump starts as soon as steam is admitted, being worked by a shuttle valve, which must be at either end of its stroke, and therefore open to admit steam to one end.

The water valves are of the multiple type—a number of small valves grouped in circular suction, 16, and discharge, 14, seatings, providing a large area with a small lift. Two such groups are fitted, one for the up-stroke, as shown in the figure, and the other alongside the first, but not shown, for the down-stroke, delivery. The valves are milled out of solid metal, and, although standardised as duplicates of each other, each valve is marked to correspond with its own particular seat. It is necessary when working at very high pressures, say above 300 lb. per square inch, to keep the valve faces in good condition and free from small leakages, because the slightest leakage destroys the faces. The maximum lift is $\frac{1}{4}$-inch, and light springs with about $\frac{1}{8}$-inch compression are fitted on the delivery valves; on the suction valves no springs are, or should be, fitted. Each valve is fitted with a guard which regulates the lift. The suction seat is driven home into the casting of the valve chest. The discharge seat is loose, and rests on a flat face which forms a valve face, on which it is carefully bedded; and the bottom of it rests on the suction guard and holds it in position; care must be taken that the two faces are bedded down equally. The screwed pin in the centre of the valve box is for keeping these guards in position, and *not* for regulating the lift of the valves.

The **Bucket Plungers** (Fig. 342) are made in two parts, upper and lower, spigoted into each other. A T-shaped carrier, made up of three rings, *A*, is fitted between the two parts, *B*, and thus leaving two spaces for the specially manufactured ebonite packing rings, *C*. In destroyer practice (Fig. 342B), the bucket is made solid with two

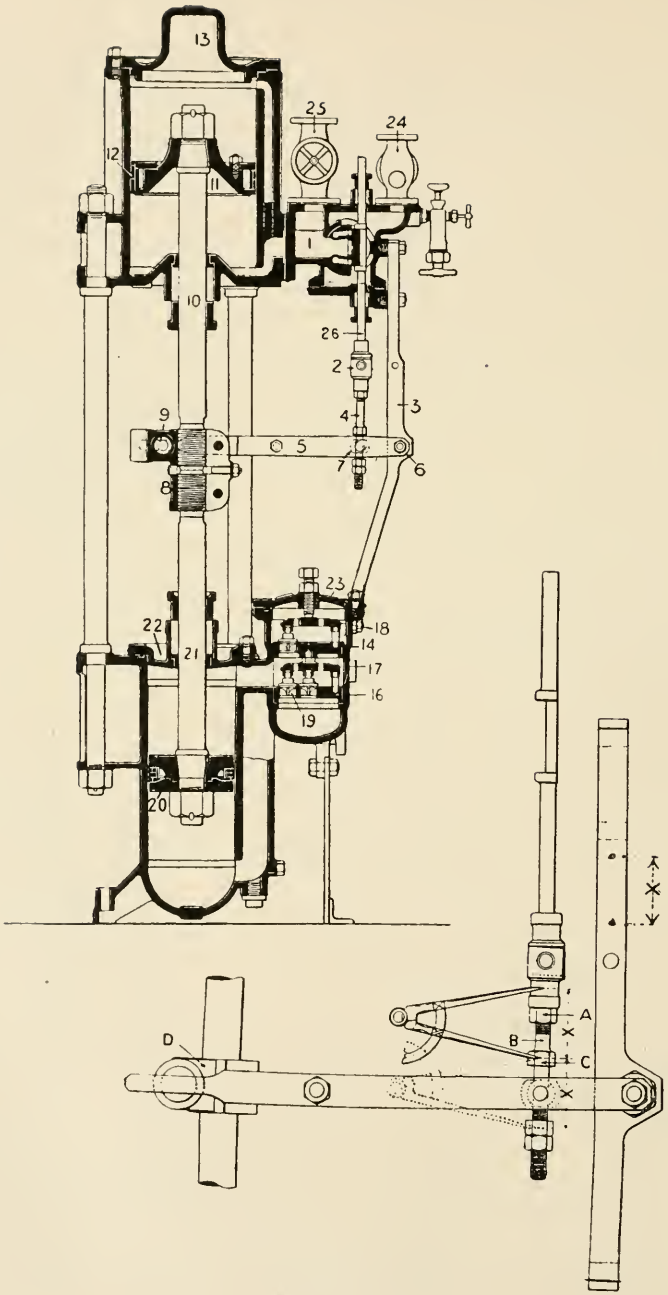


Fig. 341.—Weir Monotype Feed Pump.

grooves for the packing. The water pressure at the back of the rings keeps them up against the working barrel, and they do not necessarily require renewal, because they do not fill the pump barrel. The carrier ring should be a good fit for the barrel to prevent the ebonite rings projecting sufficiently to get broken away. The opening at the lips of the ebonite rings should be as small as possible, and there should be no vertical play in the recesses or grooves; when in place, and the bucket, large type, is screwed up, the rings should be just capable of being moved by the fingers laterally.

Before putting the ebonite rings in the grooves of the solid plunger, they should be immersed in hot water, 180° F., for about ten minutes

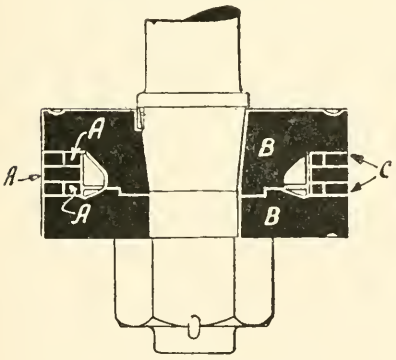


FIG. 342A.—Weir Pump Bucket.

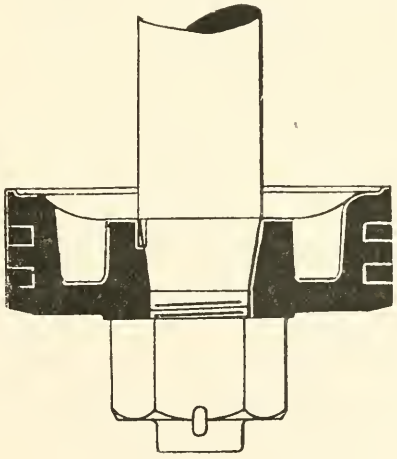


FIG. 342B.—Weir Pump Bucket.

to make them sufficiently plastic to slip over the bucket body. In pumps of large diameter, good results have been obtained by using uncut ebonite rings.

In Fig. 341 the method of adjusting the length of stroke of the auxiliary slide valve is shown, by using a pair of dividers set to a length, *X*, marked by centre dots on the front stay, and then setting the nuts to coincide. Care should be taken to lock the nuts tight after setting and not to alter the adjustment when doing so.

A series of sections and views of the Weir slide valve chest and valves is shown in Fig. 343. The auxiliary valve *B* is reciprocated by the movement of the pump rod, and it operates the shuttle valve *A*, by steam only. The shuttle valve itself is practically a D slide

valve with a circular face, and the smaller auxiliary valve works on its back.

When the piston is at the bottom of its stroke the main shuttle valve is in the right-hand position, as shown in the left-hand top view,

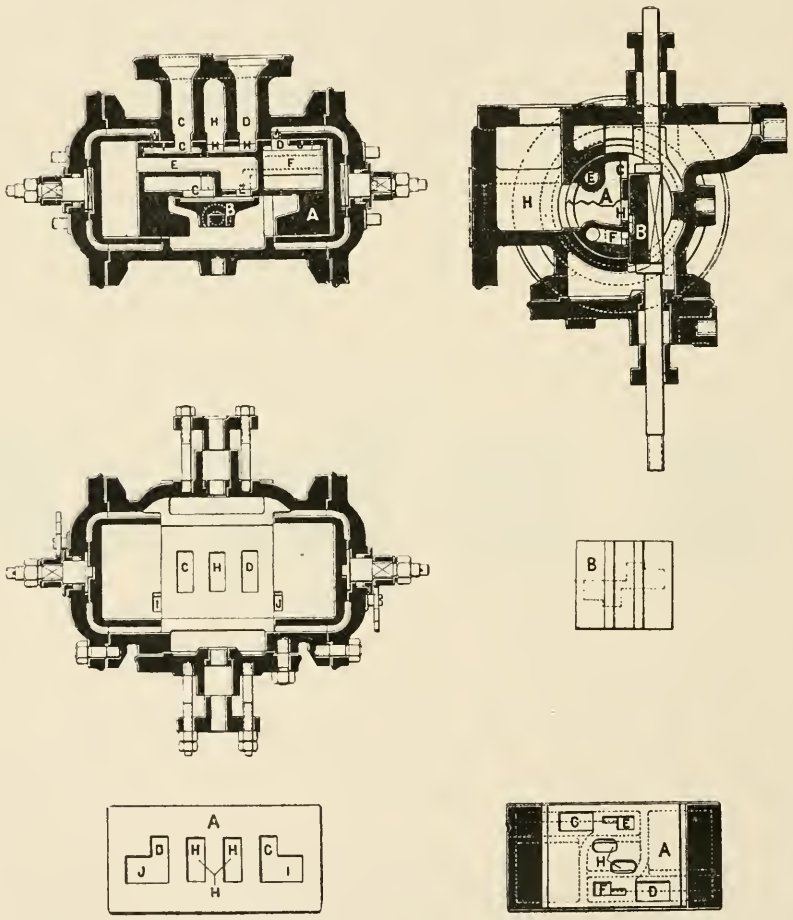


Fig. 343.—Details of Weir Feed Engine.

and the port C, leading to the bottom of the steam cylinder, is open to steam pressure, and C remains open until the piston reaches about half-stroke ; at this point the auxiliary valve B begins to move in the same direction as the piston. At about $\frac{3}{4}$-stroke the auxiliary valve B closes the port C, through the shuttle valve, and the remainder of the

piston stroke is completed by expansion of the steam already in the cylinder (or by more steam being admitted through the bye-pass, which will be described later). At the end of the piston stroke the auxiliary valve opens the port *E* leading to the left-hand end of the shuttle valve, the other end of which is open to the steam through the port *F*, and thus the main shuttle valve is thrown over until the exhaust steam from the left-hand end of the shuttle is cut off; this exhaust acts as a cushion and prevents the shuttle from hitting the end cover. The port *D* is now open to steam, and the piston begins to move on the down-stroke.

When starting with the cylinder cold, the pump will not complete its stroke by expansion of the steam only, because of condensation,

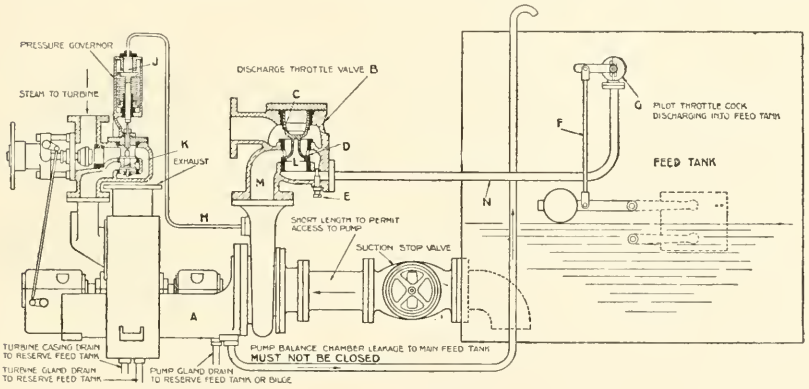


FIG. 344.—Weir Turbo-Feed Pump. General Arrangement.

and it is necessary to admit steam after the auxiliary valve closes the main ports *C* or *D* on the face of the main valve. *Bye-passes*, *I* and *J*, are made by cutting a port on each of the bells and corresponding ports on the back of the main shuttle valve. When not required, that is, after thoroughly warming through the steam system of the engine, the bells can be slightly rotated by hand at either end as necessary and the ports closed. Priming has the same effect as initial condensation in this instance, and the arrangement of the bell ports are then also useful in practice.

Weir Turbo-Feed Pump.—The general arrangement for a marine installation is shown in Fig. 344, in which *A* is the turbo-pump discharging through the throttle valve. The valve chest *B* is provided with a valve *C*, rigidly connected with a piston *D*, and it has a bye-

pass passage controlled by a small valve E. In the feed tank there is a float gear F, connected with a pilot throttle valve G, which *closes* with a rise of water-level.

When the water-level falls G opens, and the water through the pipe N discharges into the tank; the pressure below the piston L falls below that in M, and partially closes the discharge valve C, thus throttling the discharge from the pump and correspondingly increasing the pressure, which is communicated to the pipe H and acts on the spring-loaded piston J, which partially closes the steam throttle valve K, and thus slows down the turbine.

Conversely, when the water-level in the tank rises and the pilot throttle valve G is closed, the pressure in the chamber M is com-

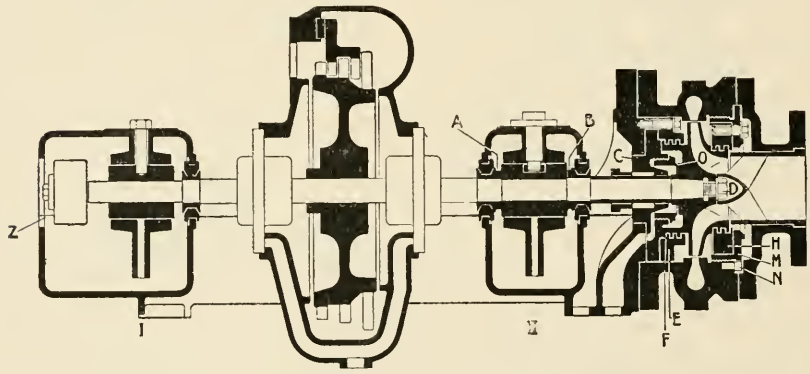


FIG. 345.—Weir Turbo-Feed Pump. Sectional Elevation.

municated to the chamber L, by means of the bye-pass E, and the piston is in equilibrium. The valve C then opens, thus reducing the pressure on the piston J, and consequently the throttle valve K opens, and more steam is supplied to the turbine in order to deal with the larger quantity of water required by the boilers. The variation in discharge pressure necessary to control the pump is so small as to be negligible in practice.

A cross section through the steam turbine is shown in Fig. 345, which is fitted between two main bearings, and the centrifugal feed water pump is fitted on the end of the overhung shaft.

The turbine is of the impulse type with one pressure and several velocity stages, two in this instance. The casing and bearings are divided horizontally, the lower frame being embodied in the general framework or base of the set, extended at one end to form a flange,

to which the pump casing is securely fastened. The turbine packing glands are fitted with carbon packing, the shaft bearings are of the self-oiling type, and one of them is used for adjusting the longitudinal position of the rotors, steam and water, and also when starting and stopping for taking any difference of end thrust. When working at or about full power there is no appreciable end thrust which is self-balanced between the two rotors. The working clearance at *A* and *B* is about $\frac{1}{1000}$th of an inch.

In the later design of pump, the impeller is fitted with two labyrinth packing rings, *H* and *E*; *H* is secured to a screwed adjusting flange *M*, with a locking pin *N*, and *E* is secured by set screws. In opening up the pump, the casing or cover forming also the suction pipe must first be removed, then the adjusting flange *M*, and next the pump casing, which exposes the set screws of the packing ring *E*. The screw *D* has a *left-hand* thread, and is marked accordingly, and after inserting a special starting screw, the impeller can be withdrawn with the packing rings, taking care that the set screws are not interfering with the removal of the parts. Finally, the front cover *C* can be removed if necessary.

The pump stuffing-box is loosely packed with graphited asbestos only. The instructions issued by Messrs. Weir should be carefully observed, and if these are intelligently carried out, there should be little trouble and good efficient working. The economy of operation is greatly improved, as with all small turbines, by the use of the exhaust heater (Fig. 336). Weir Turbo-feed pumps are constructed to operate at any pressure ordinarily used in practice, and have capacities varying from 7000 to 35,000 gallons per hour, which for a steam consumption of 14 lb. per S.H.P.-hour would provide for 5000 to 25,000 S.H.P. controlled by one unit.

Brown-Boveri Turbo-Feed Pump.—This turbine pump is made in both the horizontal (Fig. 346) and vertical types, which are similar except for the lubricating arrangements. The pump body can be turned round by steps of 45 degrees to obtain the best arrangement of the discharge pipe. There is no baseplate or coupling, and only two bearings, one of which is lubricated by the feed water, and the other (a combined thrust and supporting bearing) is, in the case of the horizontal pump, oil lubricated by two independent means, either of which alone effects efficient lubrication. In the case of the vertical type, the lubrication is effected through helical grooves drawing oil from a well under the bearing.

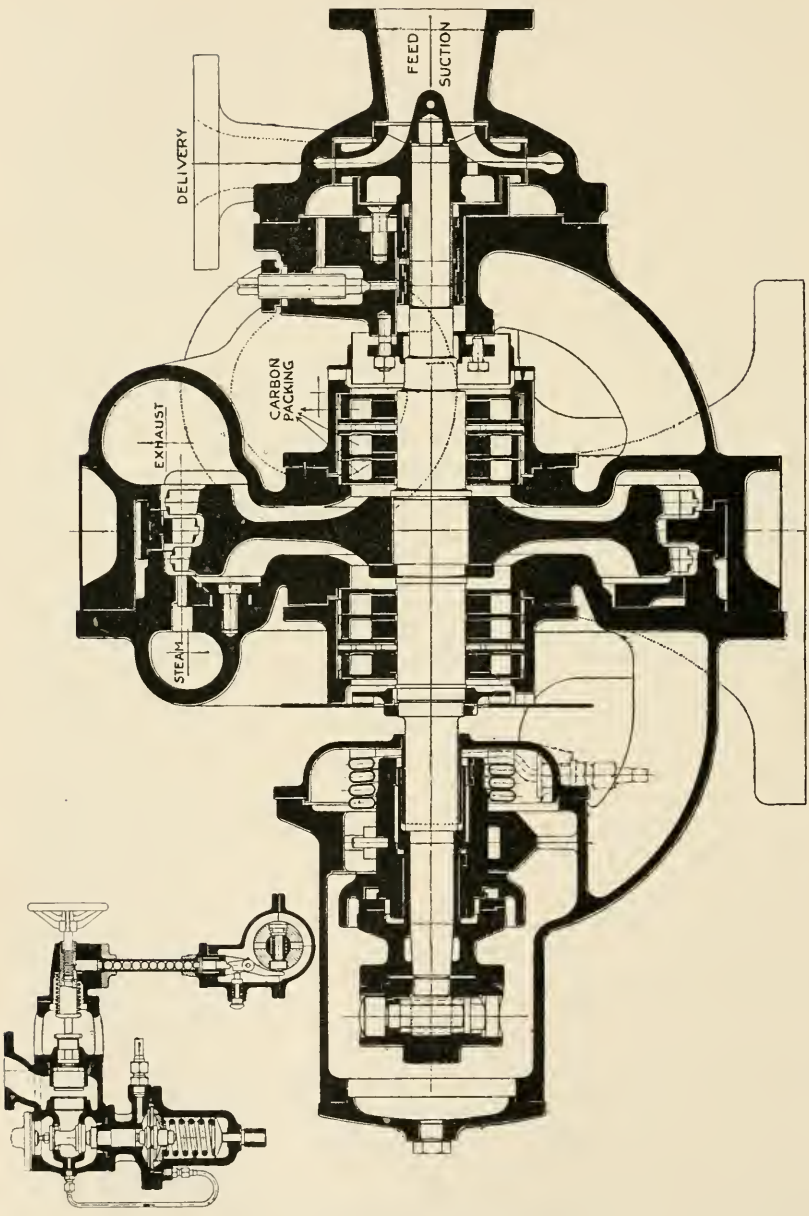


Fig. 346.—Brown-Boveri Turbo-Feed Pump.

An excess speed governor of the trigger type is fitted to operate the main stop-valve, which is quite independent of the special regulating valve.

These pumps are designed in various standard sizes, running from 5000 to 7650 revolutions per minute, and ranging from a capacity of 9000 to 22,500 gallons per hour, and, as in all cases of feed pumps, their actual capacity is much in excess of their rated capacity, as stated above. From a steam consumption curve of trial results from some of these pumps, it appears that when delivering respectively 9000 and 22,500 gallons per hour, about 75 lb. and 625 lb. of water are pumped for each 1 lb. of steam consumption. In these tests, the steam pressure was 170 to 200 lb. absolute, back pressure 17 lb. absolute, and pump discharge pressure 227½ lb. per square inch.

The feed pump is a necessary component part of any steam plant, and it would appear from the above figures that its working accounts for $\frac{1}{75}$ to $\frac{1}{62 \cdot 5}$, or for 1·33 per cent to 1·6 per cent of the total steam consumption of the plant; this is not a large proportion, considering that the feed engine (turbine) was working non-condensing, and therefore without the beneficial results obtained by a good vacuum in turbine practice.

Feed Heaters.—The object of heating the feed water is to promote economy by utilising some of the heat otherwise wasted in the condenser, and thus reducing the work of the boiler. The following may be considered the advantages of feed heating:

1. Economy, which may be anything between 1 and 20 per cent, according to the efficiency of the system employed. (The economiser is probably one of the most efficient types of feed heater; but the feed heater, as generally understood, derives its heat from steam, not from the waste products of combustion.)

2. More even temperature in the boiler, and a consequent saving from the racking strains to which it is subject, especially when raising steam and cleaning the fires.

3. Reduction of priming, and a better quality of steam generated in the boiler; it is in this way that the economiser effects part of its economy.

4. Steam can be more easily generated with hot feed water, and the heating surface appears to be more efficient.

Steam feed heaters may be divided into two classes: the injection type, which can only be used on the suction side of the feed pump; and the surface type, which can be used on either the suction or delivery side.

With *jet injection feed heaters* some of the exhaust steam (Fig. 336) from any convenient part enters the feed tank by a pipe fitted with some kind of mixing apparatus, to allow the steam to condense and mix with the water without noise or inconvenience from local heating. The heating steam is usually obtained from the coil drain or gained steam of the evaporator, from the general steam drain system, from the auxiliary exhaust pipe, or from the receivers of the main reciprocating engines (the L.P. preferably), where it has already performed some, if not the whole of the work that can otherwise be obtained from it.

The condensing of the steam amongst the feed water raises the temperature of the latter, and the limit of a jet heater is when the temperature rises to about 180° ; above this limit there is a water loss from escaping vapour, the feed pumps may be troublesome, owing to the presence of vapour and air (unless care is taken to allow them to escape) causing irregular working by their expansion when the pump draws in a fresh supply of water. All pumps work best when the water is freed from air and vapour, and, when there are any air pockets, it is good practice to connect them with the exhaust steam pipe by a small pipe and cock, and thus allow the air and uncondensable vapour to escape without losing the water carried over with them.

With *surface feed heaters* a much greater temperature can be imparted to the water by fitting them between the feed pump and the boiler, and when steam is used for heating this temperature is only limited by that of the heating steam. The supply of heat may be taken from any of the parts mentioned for jet heaters, and in a few instances is taken directly from the boiler itself.

Live-steam Feed Heater.—When the feed water is heated by steam taken directly from the boiler, without doing other work on its passage, it is termed a live-steam heater.

Two views may be taken of its action in effecting economy—either that it is conducive to the production of a better quality of steam in the boiler, and this promotes economy in the engine ; or, that the proportion of the steam used in heating the feed water is more reproductive when used in this way.

In the first case, in some particular instances, no economy could be produced by its action if the steam generated without the heater were of the highest quality, that is, perfectly saturated and dry. Some experiments tend to prove this contention, and that practically no economy was effected by feed-heating as demonstrated by them. There is, however, a mass of evidence to prove that economy has been

obtained in a great number of other cases, probably where priming was prevalent.

Direct contact heating is fitted in some turbine-driven destroyers, in which a hot-well pump delivers the feed water through the heater from which the feed pump takes its suction.

A diagram of a somewhat similar system, but with a surface heater, is shown in Fig. 336, as fitted in connection with turbines and a Weir wedge-shaped condenser.

Weir Multi-flow Surface Feed-water Heater. — Fig. 347 illustrates this heater, which may be fitted at any convenient position between the feed pump and the boiler. Steam from either the drain or exhaust system enters at the top, and is deflected to surround the heating tubes; a drain is fitted to carry away the water of condensation, and a water-level gauge is fitted to show the level, and when working the level should be just visible in the glass.

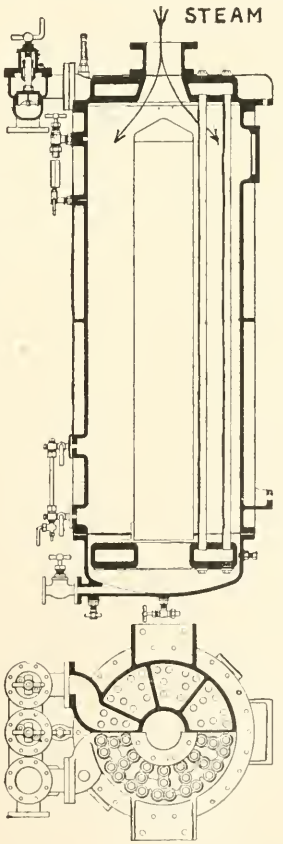


Fig. 347.—Weir Multi-flow Surface Feed Heater.

The feed water enters from the pump through the left-hand valve, shown as top of the three in the lower figure, and passes downward through the first section and then upwards, and again downwards in the second section: this is repeated several times, and finally the heated water passes out through a non-return valve to the boiler. When getting under way or starting, the heater is shut off by closing either the steam or the drain valve. If the heater is not to be used because of cleaning or other cause, the inlet valve can be closed and the water passed direct to the boiler through the middle of the three valves and over the top of the non-return valve.

The heating tubes are of copper, 12 W.G., or 0·104 inch thick, or 2·642 mm. thick, securely expanded into cast-iron tube-plates of substantial section. The holes through which the tube ends are expanded are afterwards closed

by screwed plugs. It should be noticed that the tubes have free room for expansion at all times, as the lower chambers are not rigidly connected, and form a floating chamber suspended from the tubes. The tubes can be removed and examined by withdrawal through the upper end of the heater without disconnecting any of the fittings or disturbing individual tubes.

CHAPTER XXVI

EVAPORATORS AND DISTILLERS

General Principles.—The evaporator is used to obtain fresh water from sea-water, and thus to make up the loss from leakage of the boiler feed water, which, as already pointed out, should be fresh water when used for high-pressure boilers. Water must also be supplied for drinking, cooking, and other purposes; this is also obtained by using an evaporator, and, in conjunction with it, a condensing and cooling apparatus, called the distilling condenser.

By using an evaporator the scale, otherwise deposited on the boiler heating surface, is deposited in the evaporator, principally on the heating coils; the boiler heating surfaces can therefore be kept free from scale and in a high state of efficiency for long periods.

The evaporator is now generally vertically arranged, as shown in Fig. 348 (*Kirkaldy*), as it then takes up less ground space. The chamber is nearly cylindrical for a great part of its height, and in its lower part coils of copper piping are fitted, which are nearly immersed in sea-water, admitted, or pumped, into the evaporator as required. *Primary steam* is admitted to the coils internally, and the sea-water is boiled and *gained steam* is produced by the heat thus imparted; while at the same time the steam inside the coils is condensed and continuously drained away through the *coil drain* to allow fresh steam to take its place.

The *capacity of an evaporator* is measured by the amount of heating or coil surface it contains; but its rate of production (which might also be considered its capacity) depends on the difference in temperature between the heating or generating steam and the generated or gained steam, and on the cleanliness of the heating coils, both internally and externally.

The same principle is used in all evaporators, and it is only the difference of arrangement of the details that constitutes any difference

in working. The mountings are similar to those on a boiler, in which the heating coils take the place of the furnace. Sea-water is admitted nearly up to the top level of the tubes, as shown in Fig. 348; steam is admitted inside the coils, and heats the water surrounding them, and finally ebullition takes place. The steam thus generated is called

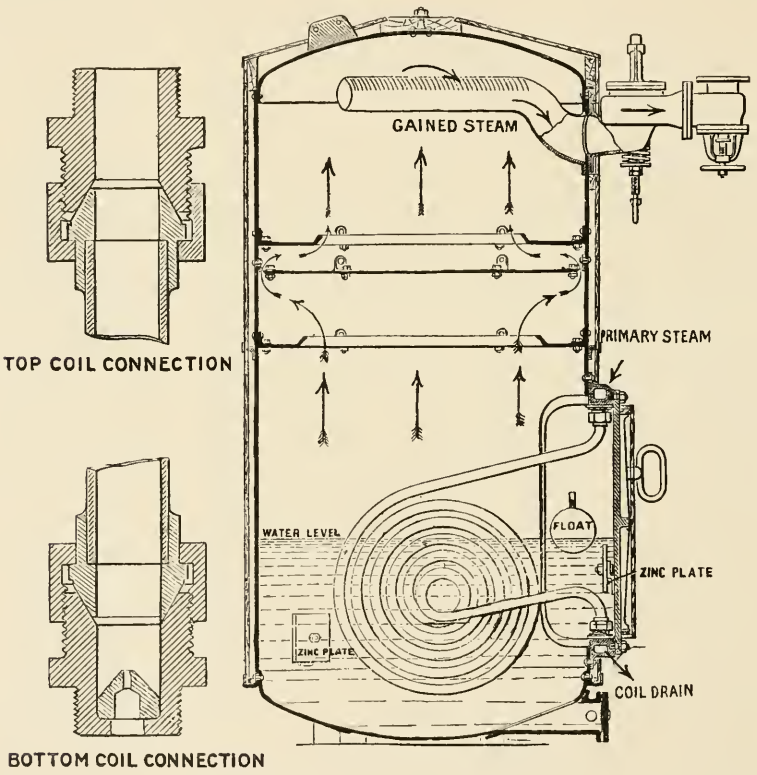


Fig. 348.—Section through (Kirkaldy) Evaporator.

the *gained or secondary steam*; and the steam used inside the coils, which may be taken either from the boiler direct, or from the main engine receiver, or from the closed exhaust system, is called the *primary or generating steam*. The gained steam is taken away from the top of the chamber through a valve and pipe, either connected with the condenser, or, as in a few instances, with the L.P. receiver, where it supplements the exhaust steam from the M.P. cylinder. The water condensed in the coils, by parting with its heat in generating the gained

steam, is led away by a drain-pipe, called the *coil drain*, into the condenser, feed tank, or feed heater ; whence it is returned to the boiler, with practically little loss from leakage. When the evaporator is used for making water for drinking purposes, the gained steam is taken to a separate condenser, called the *distilling condenser*, from which it runs or is pumped into the ship's tanks set apart for the purpose.

Evaporator Fittings and Mountings.—The fittings and mountings on an evaporator are very similar to those fitted to a boiler, and are for very similar usage. They are generally as follows :

FIGS. 348 TO 357

Safety valve for generated or gained steam.
Pressure gauge for gained steam.
Outlet steam control valve, or vapour control valve.
Outlet stop valve.
Inlet for low-pressure steam to coils, non-return valve.
Inlet for high-pressure steam to coils.
Drain valve from coils, conveying away water condensed from the steam in the coils.
Safety valve in connection with coils.
Pressure gauge in connection with coils, primary steam.
Water gauge.
Blow-out valve connected with the sea.
Blow-out valve connected with the bilge.
Feed inlet and regulator. (Inlet to regulator fitted with a sentinel or safety valve.)
Brine mixing and cooling chamber.
Cooling water inlet to brine chamber from feed-supply pipe.
Brine suction from cooling chamber to brine pump.
Hydrometer fittings for testing density.

Starting and Working the Evaporator.—All evaporators prime readily, from the same causes which induce priming in a boiler. Irregular working, leakage of the internal feed pipe above the working level, dirty water, soda or potash mixture, the impact of cold air on the outside of the steam space when imperfectly lagged, and the low pressure at which the steam is generated, are the principal causes in ordinary practice.

The necessary openings of the valves and pressures to obtain the full output of the evaporator are generally shown on an instruction

plate attached to the front. The working level is generally at or below the upper tubes, and if the water gauge is correctly fitted it should then show nearly a full glass.

When starting, it is advisable to proceed in the following order :

Adjust the brine index, if required.

Start the pump and fill the evaporator until the feed regulator exerts a pressure.

Open the coil drain, generally *full open* when working.

Admit steam to the coils *slowly*, and allow the pressure to rise inside the evaporator steam space to 2 or 3 lb. above the atmospheric ; then very gradually open the vapour outlet valve until the requisite pressure or vacuum is obtained.

The guaranteed output, when clean, can be obtained either with a pressure of about 6 lb. in the coils, and 9 inches of vacuum in the steam space, or with 16 lb. in the coils and 1 to 2 lb. pressure in the steam space.

Test the density every hour, and see that it agrees with the brine index. If it does not agree, there is generally some unnecessary leakage.

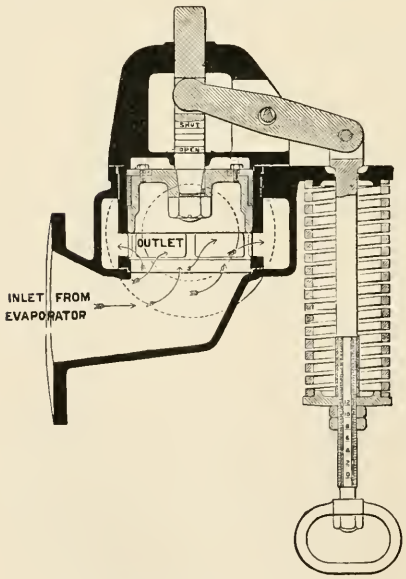


Fig. 349.—Vapour Control Valve.

The evaporator should be blown down every twelve hours. To do this, close the vapour valve and allow the pressure to rise to about 12 to 15 lb. per square inch; then open the blow-down quickly to the sea, and as soon as the water is entirely blown out, close the inlet steam valve to the coils. The entrance of the cold feed water creates a vacuum, and water is drawn in quickly through the blow-out, which should be closed as soon as the water begins to show in the gauge glass, when the inlet can be opened and the working restarted.

Vapour Control Valve.—Fig. 349 shows a cross section through a Kirkaldy vapour control valve. There is a cylindrical sliding valve working inside a cylinder. Below, the valve is connected with the steam space of

the evaporator through an internal steampipe. Above, the valve is open to the atmosphere, which tends to force the valve downwards and so cover the outlet ports. The valve is packed so as to ensure a good working fit in the barrel; and a ring face is formed on its upper surface which, when the valve is fully open, makes a steam-tight joint with a seating on the cover. A spring, in compression,

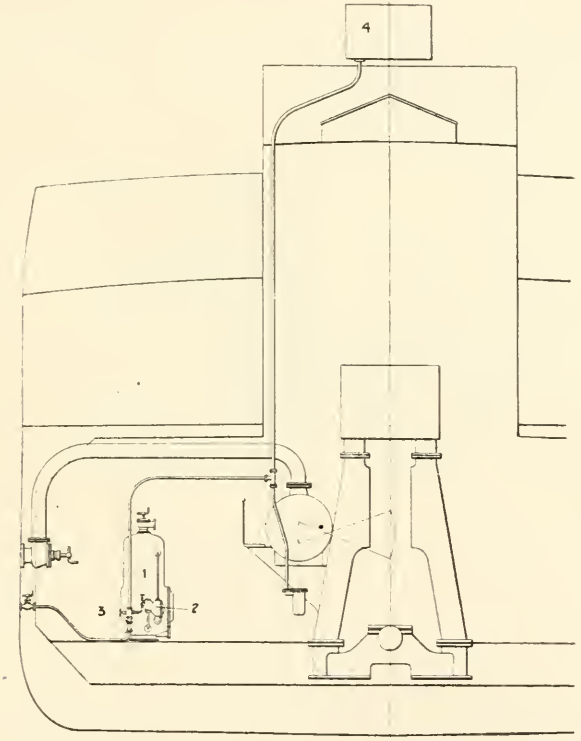


FIG. 350.—Weir Mercantile Evaporator Plant.

tends to keep the valve open in opposition to the atmospheric pressure, and by adjusting the compression of the spring, the valve can be set to maintain a constant pressure or vacuum in the evaporator steam space. In the figure the valve is shown fully open, and the spring set for 8 inches vacuum by means of check nuts.

Weir Mercantile Evaporator Plant. — Fig. 350 shows the general arrangement of the relative positions of the (1) evaporator,

(2) feed regulator, (3) brine ejector, and (4) sanitary tank, which can be used instead of a brine pump to obtain continuous brining and a regular density of the water in the evaporator. The sanitary tank is fitted at a sufficient height to obtain the necessary head to permit automatic feeding through the feed regulator and at the same time operate the brine ejector for discharging the brine overboard. The supply to the ejector and tank is obtained from the sanitary pump worked off the main engine, as shown, or from an independent pump.

Fig. 351 shows a sectional view of the brine ejector and valves. The brine valve is designed so as to provide for fine regulation to maintain a regular density of between $\frac{2}{32}$ and $\frac{3}{32}$.

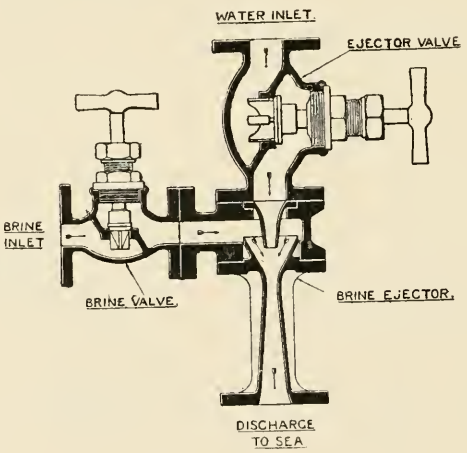


Fig. 351.—Weir Brine Ejector.

Fig. 352 shows the feed regulator, consisting of a copper float ball operating a cylindrical valve in a suitable casing in which ports have been cut to correspond with those in the valve. When open, the valve admits water through a check valve which, when the evaporator is working, is kept full open. The regulator maintains a constant water level in the evaporator at any rate of output up to the maximum for which the evaporator is designed.

Fig. 353 shows another form of feed regulator as fitted in warships. By raising or lowering the handle H, which raises or lowers the fulcrum of the float lever, the water level in the evaporator can be varied. The feed water enters at A, from the feed pump, and passes the floating valve B into the passage C to the evaporator. When

the float E rises to the working level, it closes the control valve F, and the pressure rises in the space below the valve B, and raises the valve which is connected through a pipe G, with the pump side of the check valve. When B rises, it cuts off the supply of feed water through C. When the float falls the valve F is opened, relieving the pressure below B which is opened by the excess of pressure above it, and admits feed water to the evaporator.

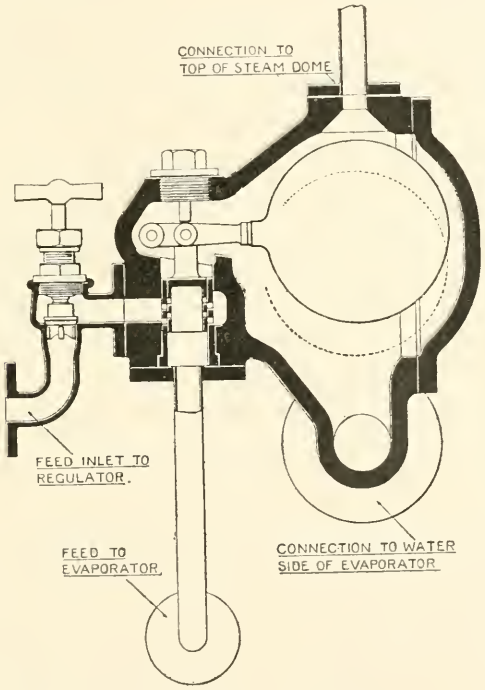


FIG. 352.—Weir Evaporator Feed Regulator.

A general arrangement of the evaporating and distilling plant as fitted in destroyers is shown in Fig. 354, and the reference gives the names and uses of the various component parts.

Fig. 355 shows in detail the arrangement of combined fresh water pump B, circulating pump A, and brine pump C. With this arrangement a spring loaded valve, 28, in Fig. 354, is fitted on the circulating outlet from the distiller, to obtain sufficient pressure for feeding the evaporator through a connection which is fitted with a pressure gauge, 24. The brine pump C has a suction connection, 13, to the evaporator,

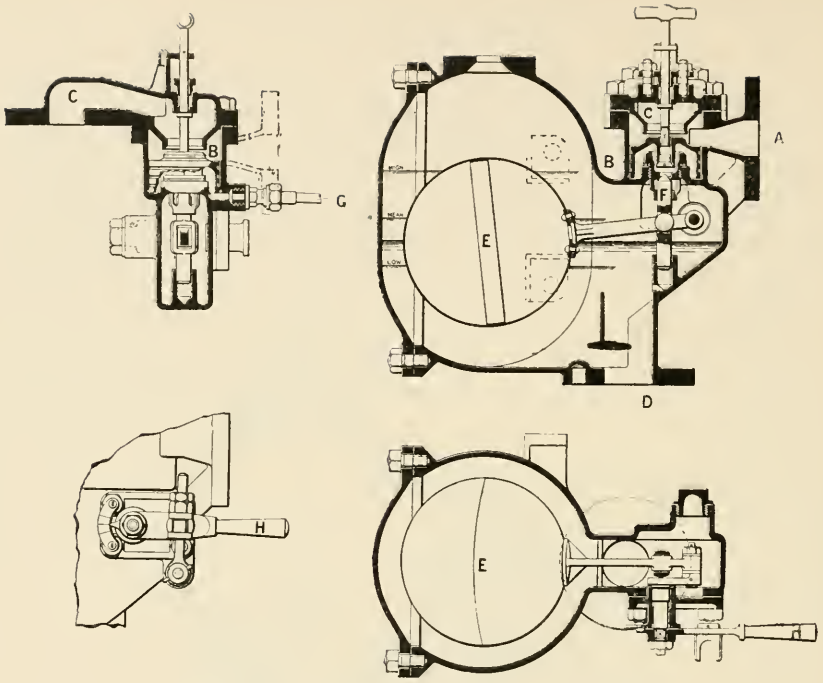


Fig. 353.—Weir Evaporator Feed Regulator for Destroyers.

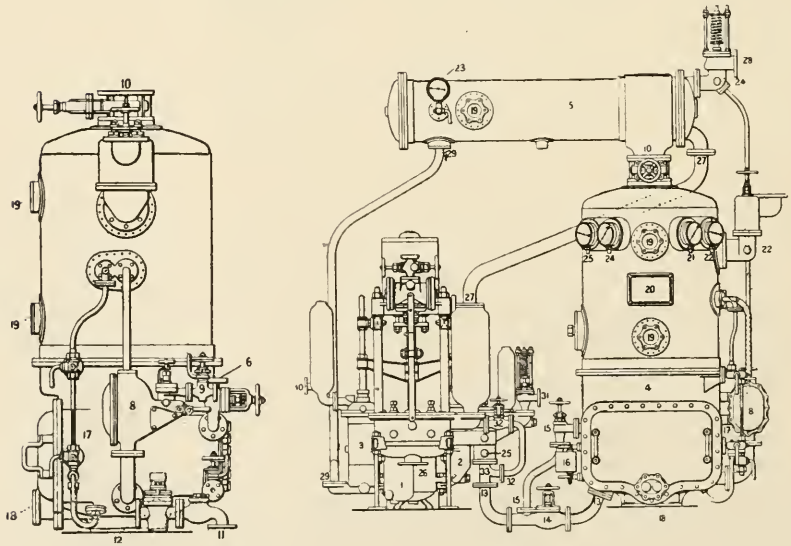


Fig. 354.—Weir Evaporator and Distilling Plant, Destroyer Type.

Reference to Fig. 354.

1. Circulating pump.
2. Brine pump.
3. Fresh water pump.
4. Evaporator shell.
5. Distilling condenser.
6. Steam inlet to coils.
7. Drain to coils.
8. Feed regulator chamber.
9. Feed check valve.
10. Generated steam outlet to distiller.
11. Blow-down cock.
12. Blow-down pipe.
13. Brine suction pipe.
14. Brine suction valve.
15. Upper brine suction.
16. Salinometer cock.
17. Water gauge.
18. Cleaning doors.
19. Cleaning doors.
20. Instruction plate.
21. Pressure gauge, inlet steam to coils.
22. Combined gauge, generated steam.
23. Combined gauge, distiller.
24. Pressure gauge, circulating discharge.
25. Pressure gauge, brine discharge.
26. Circulating inlet.
27. Distiller circulating outlet.
28. Safety valve on circulating outlet.
29. Distilled water inlet to pump.
30. Distilled water discharge from pump.
31. Brine discharge.
32. Brine diluting valve and pipe.
33. Brine diluting connection.

controlled by a brine valve, 14, and a brine scumming valve, 15, and is connected with the circulating pump discharge by a valve and pipe, 32, for diluting and cooling the brine. The system requires regulation

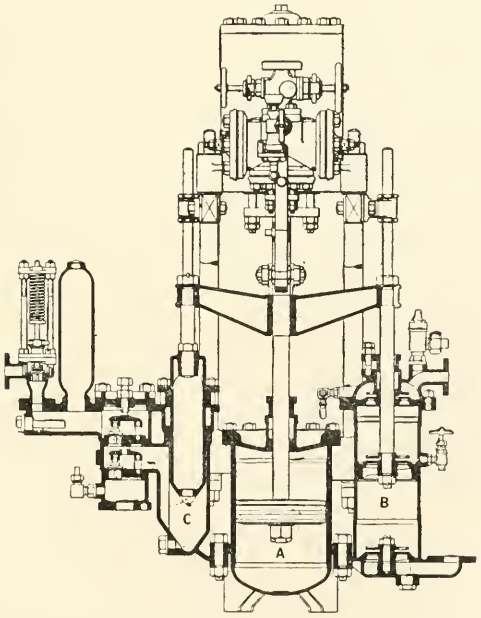


Fig. 355.—Weir Fresh Water, Circulating, and Brine Pumps.

for a constant supply of cooling water to maintain a constant density, because the supply of cooling water naturally affects the amount of brine withdrawn by the pump.

Morison Evaporator.—In Fig. 356 a Morison evaporator is shown with group of coils removed for cleaning. It consists of a vertical cylindrical chamber provided with necessary formations for fitting several coils in its interior. The lower part is fitted with a swing door on a pivoted arm, for convenience of ready removal, and when open the coils are exposed. The heating coils are arranged in threes, and are detachably connected with two tubes, forming respectively the inlet for steam and the outlet for the resulting water condensed. The attached coils can be readily swung and withdrawn from the vessel for cleaning purposes, and afterwards swung back again. When it is necessary to replace a set of coils with a clean set, the spare set is lifted on and fitted up. After the coils are connected in their working position, steam can be turned on and all the joints tested in place before the door is closed up: this is a great convenience, and any defective joints can be made good.

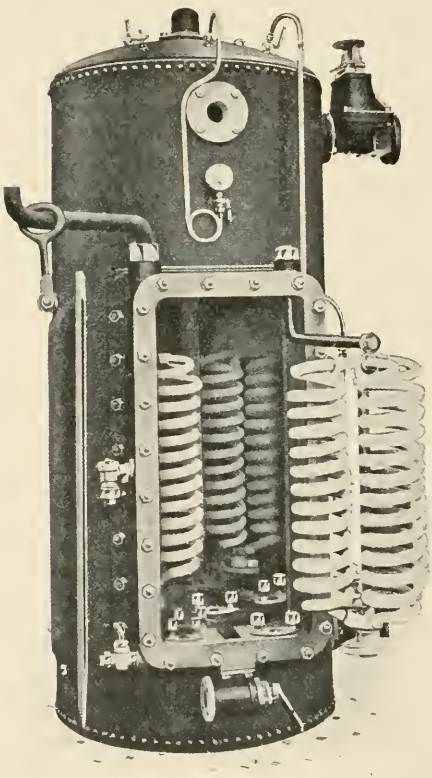


Fig. 356.—Morison Evaporator.

Kirkaldy Evaporating and Distilling Apparatus.—A section through a Kirkaldy evaporator is shown in Fig. 348; and Fig. 357 shows one complete set of two evaporators *A* and *B*, one feed-water heater *C*, one brine pump *D*, one distilling condenser *E*, and one circulating and one fresh-water pump *F* (driven by the same engine), as fitted in each of the turbine engine rooms of the *Dreadnought*. Each evaporator can be worked separately, and the gained steam either passes away through the vapour valves to the condenser, or through separate valves connected with a common pipe leading into the distilling condenser from which it is removed, as water, by the fresh-water

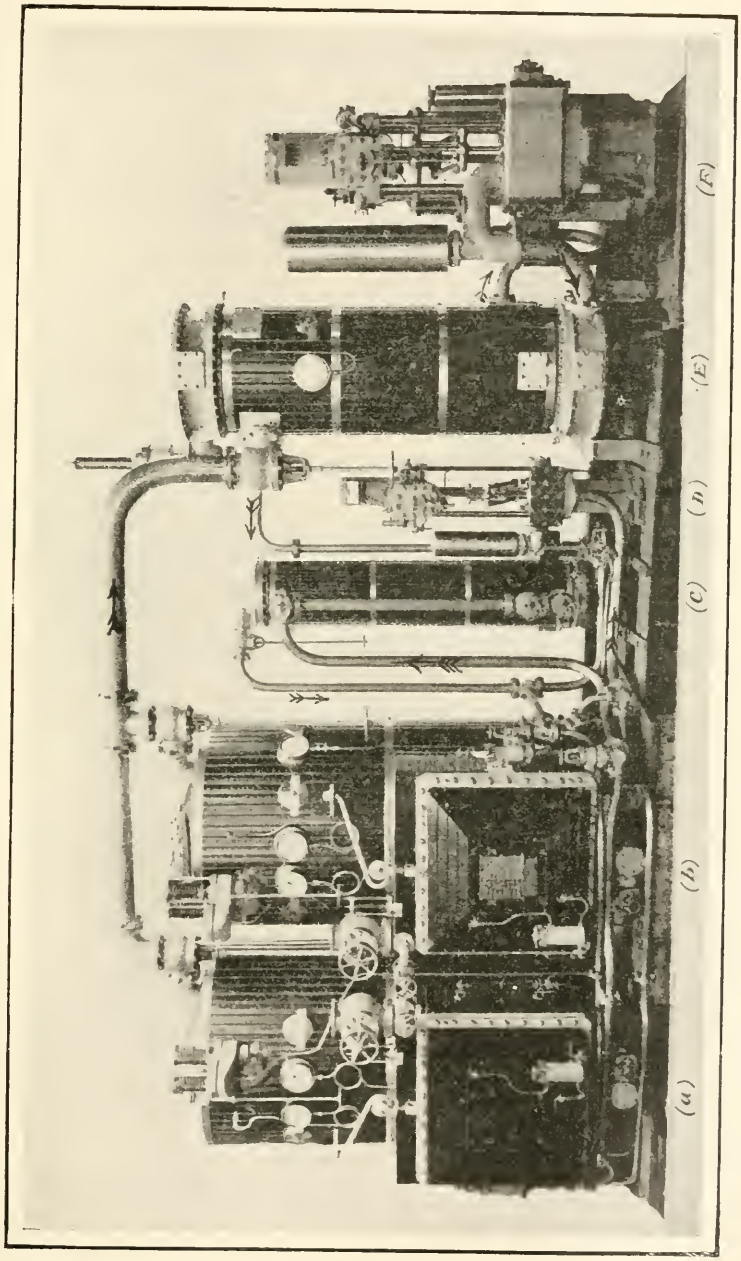


FIG. 357.—Kirkaldy Evaporator and Distilling Plant.

pump. The condensed primary steam from the coil drain passes either into the feed-water heater, if required, or into the feed tank or condenser through a bye-pass pipe. The brine is removed, through a diluting box (brine cooler), from the lower part of the evaporator by the brine pump and discharged overboard. The evaporator feed water is supplied by the circulating pump ; it first passes through the distiller and, becoming slightly warmed during its passage, is discharged into the bottom of the feed heater ; it next passes upward through the feed heater and is discharged at the top through the check valve into the evaporator feed pipe. A part only of the circulating water may be required as evaporator feed, and the surplus escapes overboard through a spring-loaded valve, shown near the top of the distiller. Each of the evaporators will produce, when clean, about 50 tons of fresh-water steam from sea-water in 24 hours' working ; and the total production for the four evaporators is thus 200 tons, or nearly 45,000 gallons, in every 24 hours.

In the Kirkaldy distilling condensers and feed heaters, the coils are of elongated spiral form ; some of the early evaporators were also fashioned in the same way. The advantage of a spiral formation is that a large area of heating surface can be put into a small space, and generally that freedom for expansion under variations of temperature which must be allowed for in some way.

Fuel used for Producing Fresh Water.—Before the introduction of the evaporator, water was produced by condensing steam taken directly from the boiler, and about 7 to 8 tons of water were produced per ton of coal burnt. The scale left on the heating surfaces reduced the efficiency of the boiler considerably, and the evaporator, when using boiler steam for which an average of only 5 tons of fresh water is produced per ton of coal burnt, by maintaining the boiler efficiency, repays the consequent loss compared with direct distillation.

If the steam used for the evaporator be taken from the closed exhaust system, otherwise wasted in the condenser, the average production is increased to 10 or more tons ; on the other hand, there is an increase in the wastage of boiler feed water.

There is little room for improvement in the thermal efficiency of the evaporator, because nearly 90 per cent of the heat supplied to it may be used in producing fresh water, and for any economy we must therefore look to the source from which the supply is obtained. For warships, which are distilling when in harbour in many parts of the world, where the shore supply is not above suspicion, the steam supplied

for heating the coils can be taken from the closed auxiliary exhaust system. There is another system, known as the "multiple" or "double-effect" system, which appears applicable to the special case of warships, and which had been applied with some success in some of them anterior to the application of the closed exhaust system.

The system of *double-effect evaporators* consists in fitting two evaporators in series. The gained steam in the first of the series is employed as the generating steam in the second. Practically little heat is lost in the first, and any steam gained increases the output per ton of coal burnt. From 10 to 12½ tons of water have been obtained per ton of coal in this way. The objection to the use of "double effect" is *priming* caused by the difficulty of regulating the evaporators when fitted in series; but this system, with the more enlightened experience now gained with evaporators worked at low pressures and fitted with a fairly efficient control valve on the outlet pipe of the gained steam, is likely to produce more favourable results, and is successfully used in shore installations.

An allowance of about 3 gallons of water is required for each person in the complement, with an extra allowance of 2 gallons for each one of the engineer staff, and 1 gallon extra for each officer, *per diem*. This works out at about 20 tons of water per day for general purposes in a battleship or large cruiser, and the boiler loss in harbour which has to be made up varies from 4 to 6 tons per day; the cost in coal is therefore about 3 tons, or a weekly consumption of about 21 tons in harbour for producing fresh water alone.

The capacity of the evaporator plant in warships is generally from five to six times the capacity of the distiller plant, and an allowance of about 10 tons per 24 hours for each 1000 I.H.P. is generally fitted for the combined output of the evaporators. A greater quantity of water could be passed through the distillers than that specified, but any increase in quantity would of course produce a corresponding increase in the temperature of the water condensed.

Other well-known makers of evaporators, distillers, and feed-heaters are *Messrs. Caird & Rayner*, the *Liverpool Engineering Co.*, *Quiggans*, *Mirlees*, *Royle* (Row's patents), and *Normandy*. The principal differences exist in the form and fitting of the heating coils, and in facilities for cleaning and repairs.

Quantity of Feed Water required to Maintain a Constant Density.—The weight of sea-water which is admitted to the boiler or evaporator as feed water is not all evaporated, because a portion is

blown out, as water, to reduce the density of the water remaining in the generating vessel.

If the brine blown out be supposed to contain n times the density of the feed water, and x = weight of steam leaving the boiler or evaporator, y = weight of water blown out; then, if the feed water contains $\frac{1}{a}$ part of salt—

$$\text{The weight of salt pumped in} = \frac{x+y}{a},$$

$$\text{and the weight of salt blown out} = \frac{n \cdot y}{a}.$$

If a constant density is maintained, the weight of salt blown out must be equal to the weight of salt pumped in, and therefore—

$$\frac{x+y}{a} = \frac{n \cdot y}{a},$$

$$\text{and } y = \frac{x}{n-1}.$$

$$\text{The weight of feed water required} = x+y = x + \frac{x}{n-1} = \frac{x \cdot n - x + x}{n-1}.$$

$$\text{The weight of feed water } (x+y) = x \cdot \frac{n}{n-1},$$

$$\text{and the weight of water which must be blown out, } y, \Big\} = \frac{x}{n-1};$$

$$\text{or, proportionately, } \frac{y}{x} = \frac{1}{n-1}.$$

For example, if $n=2$, then $y=x$
$n=3$,, $y=\frac{1}{2}x$
$n=4$,, $y=\frac{1}{3}x$, and so on.

And if $n=2$, the quantity of feed water required $=2x$
$n=3$,, ,, ,, $=\frac{3}{2}x$
$n=4$,, ,, ,, $=\frac{4}{3}x$
$n=5$,, ,, ,, $=\frac{5}{4}x$.

Heat Lost by Blowing Down.—The least scale is deposited in an evaporator when the density is kept at about $\frac{2}{32}$, or 20°, by the

service hydrometer, and, from the previous section, the quantity of feed water required is then about twice the amount of water evaporated. The amount of heat carried away through brining down can be calculated as below—

Let x = quantity of water evaporated during any given time ;
y = quantity of water blown out during the same time ;
t = temperature of water pumped in as feed ;
T = temperature of steam generated.

Then, because the brine is blown out as water at the temperature T—

$$\text{The loss of heat} = y \,.\, (\mathrm{T} - t).$$

The heat expended usefully, that is, in evaporating water—

$$= x \,.\, (1114 + 0{\cdot}3\mathrm{T} - t) ;$$

and, expressed as a fraction, the heat loss—

$$= \frac{y \,.\, (\mathrm{T} - t)}{x \,.\, (1114 + 0{\cdot}3\mathrm{T} - t) + y(\mathrm{T} - t)}.$$

If the density is maintained at n times that of the feed water—

$$\frac{y}{x} = \frac{1}{n - 1},$$

and therefore the loss $= \dfrac{\mathrm{T} - t}{1114 \,.\, (n - 1) + \mathrm{T}(0{\cdot}3n + 0{\cdot}7) - n \,.\, t}$.

EXAMPLES.—(1) If n be four times the density of sea-water, and the feed water be of the same density as sea-water, as when using a jet condenser, and $t = 60°$, T = 300°—

Then the loss = not quite 6·2 per cent.

(2) If $n = 2$, $t = 80$, and T = 220, as when using an evaporator—

Then the loss = not quite 11·3 per cent.

Cleaning Evaporators.—The rate at which steam is generated in an evaporator is due to the rate of transmission of heat from the generating steam to the water surrounding the coils, which rate is dependent on the difference of temperature between the generating and generated steam and the cleanliness of the heating surfaces, both internally and externally.

When the generating steam is taken directly from the boiler, there is only a very slight deposit of grease inside the tubes of the coils ;

but when it is taken from the engine receivers, or from the closed exhaust system, the deposit may seriously decrease the efficiency. The amount of the deposit is intimately connected with the internal lubrication of the engines; and when boiler steam is used, with the efficiency of the grease extractor. Many engineers use no internal lubrication for either main or auxiliary engines, and very few use any for the auxiliary engine cylinders; but a small quantity of greasy matter finds its way into the steam from the piston rods and valve spindles. For good practice, such as the above, the internal surfaces of the tubes may require cleaning after about six months' total working; if the external surfaces are clean, the amount of the internal deposit is evident from the falling off in the rate of production.

The internal surface of the tubes can be best cleaned by scalding them through with a strong solution of caustic potash or soda, and then cleaning them with a wire-handled brush and clean water. Generally the coils must be removed from the evaporator for this purpose, and in some cases they can be placed over a slow fire and the grease burnt off; but care must be taken not to weaken the metal of the tubes when doing this.

The external surfaces of the tubes should be scaled with a blunt instrument after about ten days' working if steam is generated below the atmospheric pressure, or after about seven to eight days if steam is generated above the atmospheric pressure. Doors are provided for access to the coils for temporary scaling, but to thoroughly clean the coils it is better to remove them from the evaporator. The scale can be easily removed if the evaporator is worked up to the moment of opening up, and just previous to this the evaporator should be blown down and refilled, to soften and crack off the scale. In no case should the scale be allowed to dry and harden on the coils, because it is then very difficult to remove. In some cases the coils can be heated up when the door is open, and a fire-hose can be used to suddenly cool them, and some scale cracked off. Care should be taken not to roughen the surface of the tubes when scaling, because any roughness increases the difficulty of detaching scale in the next cleaning.

It should be possible to thoroughly clean the heating coils and internal parts of the evaporator in one working day. A scale should be allowed to remain on the upper parts of the steam space to protect it from corrosion; but if this scale is loose, it should be removed and a thin wash of Portland cement brushed over it before closing up. Two or three coats are generally applied, and the wash is preferably

mixed with sea-water. Zinc protectors are generally fitted in the steam space, and sometimes in the lower part near the coils; but the latter is now sometimes unnecessary, because the coils are made of copper and the casing of gun metal.

After replacing the coils, the joints should be tested by turning on steam. All well-arranged evaporators allow this to be done before replacing the door; any leakage tends to induce priming.

When perfectly clean and free from scale, an evaporator will produce about 1 ton of gained steam for each 1·1 to 1·25 tons of steam condensed in the coils; but the formation of scale gradually reduces the original efficiency of 90 or 80 per cent to about 55 or 50 per cent with about $\frac{1}{8}$-inch scale, and thus increases the steam consumption per ton of steam gained by $1\frac{3}{4}$ to 2 tons.

By increasing the pressure (and, therefore, temperature) of the generating steam, the capacity can be fairly well maintained, although the efficiency is decreased by lack of cleanliness.

Distilling Condenser.—The principle of a distilling condenser is exactly the same as that of the main or auxiliary condenser; but the distiller is usually smaller, and arranged vertically instead of horizontally, not for efficiency, but for convenience.

In Weir's arrangement of distilling condenser, the tubes are vertically arranged, as shown in Figs. 358 and 359. The circulating water passes internally through them in an upward direction, and the steam from the evaporator or boiler entering from the top, being condensed on the outer surfaces of the tubes, falls to the bottom of the chamber. The condensed water is allowed to accumulate until the level rises to that shown, when it can be either run or pumped away through an outlet fitted on the air pipe at the side of the distiller. The accumulation of water allows a much greater time for cooling it as well as condensing it, and the water being a better conductor of heat than the steam from which it is condensed, the process of cooling is accelerated to the same extent. Air is admitted through the valve shown, to a space formed between a hollow cylindrical hanging screen and the outer casing. This free admission of air balances the pressure, by preventing the formation of a vacuum from the condensation of the incoming steam with that of the outflow pipe, which is also an air pipe. The air admitted mixes to some extent with the incoming steam in the distiller, and aerates it. The water produced is cold; and the outlet for the condensed water, it will be noticed, is nearest the circulating inlet, or coldest water and surface.

The tubes are expanded to make them steam-tight in their holes, as in an ordinary boiler. Expansion of the tubes is allowed by making

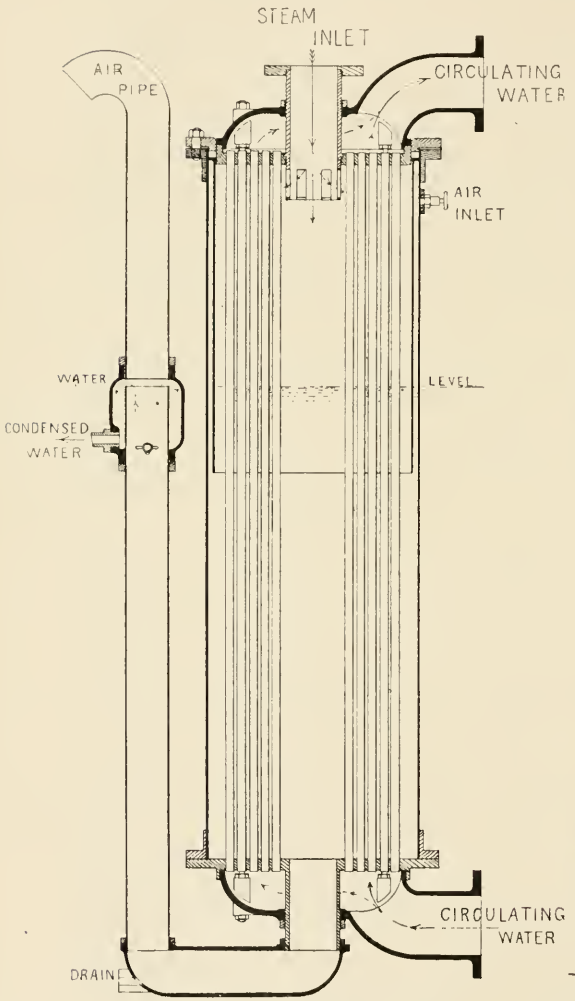


Fig. 358.—Weir Distilling Condenser.

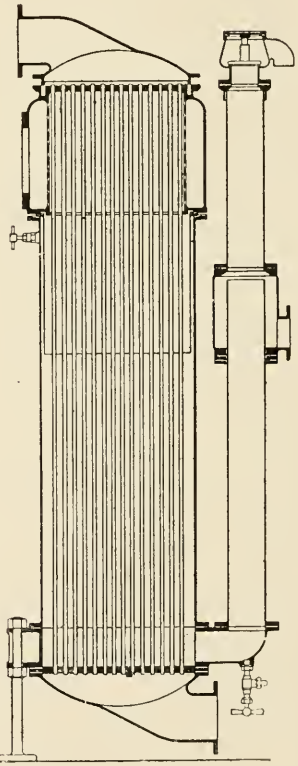


Fig. 359.
Weir Distilling Condenser, Vacuum Type.

the top tube plate in the form of an expansion gland. Any distorting effect produced is taken on bends provided in the pipes in connection with the steam inlet and circulating outlet. The top and bottom end plates, or covers, are stayed together by three or four long stays passing

through cooling tubes. A test cock is fitted on the outlet pipe for the condensed water, and there is a drain at the bottom of the same pipe.

The water from the steam gained in the evaporator, when condensed in the main or auxiliary condenser, is of a slightly greasy nature, and not suited for drinking purposes. The sea-water in the evaporator from which the steam is produced is perfectly clean, and therefore free from grease ; but the fresh water produced from it is rather tasteless, because it is freed from air and impurities in the process of boiling. By condensing it in a clean chamber, the distilling condenser, it remains free from grease, and means are provided for aerating it during condensation. In earlier times the steam condensed for drinking purposes was taken directly from the boilers, and although condensed in a separate and clean vessel, was always more or less impregnated with grease ; but since the use of the evaporator the water has generally been absolutely pure and wholesome, although slightly tasteless. The water now supplied to the boilers from the feed tank compares very favourably with that which was originally supplied for drinking, and can still be used if other sources fail.

Fig. 359 being used for condensing under a vacuum differs slightly from Fig. 358, and has a non-return valve on top of the air pipe. The principle of condensation is identical.

PART VIII

PROPULSION

CHAPTER XXVII

SHIP OR HULL RESISTANCE

When a body of wedge-shaped form is completely submerged in a fluid, and moves through the fluid in the direction of its length, the

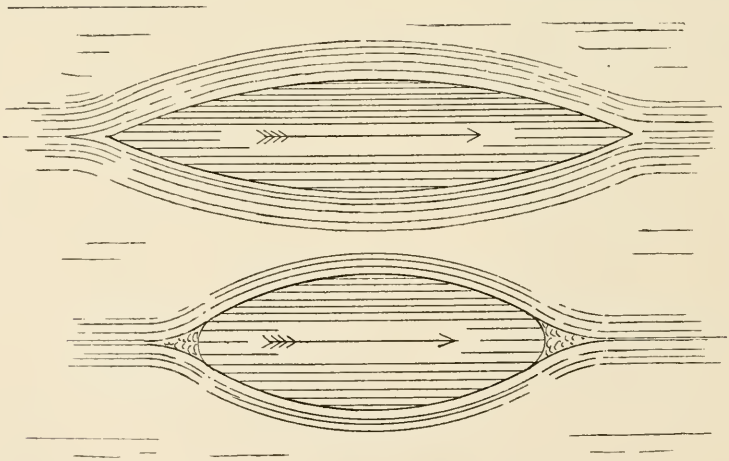


Fig. 360.—Stream Lines.

fluid opens out in front of the body, moves round it in definite lines, called *stream lines*, and closes in behind ; the pressure which is exerted by the fluid on the front of the body is nearly balanced by that behind it. These stream lines are of perfectly definite form, which differs according to the shape of the body. In all cases of ordinary fluids the

body must be of nearly oval shape, and of gradual curvature for perfect stream-line motions. If sharp corners are presented to the fluid passing round them, some energy is dissipated, and the head resistance is consequently increased by the diminution of the pressure on the stern. In the actual case of a ship moving through the *water* a considerable resistance to motion is experienced. This is due to four causes :

1. The fluid is imperfect, possessing the property of viscosity.
2. The surface over which the water passes is not frictionless.
3. Eddies are formed, because the body is not of perfect shape.
4. The body moves on and near the surface of the water and not entirely below it, and thus waves are formed.

1. **Viscosity of Water.**—When at rest, water is almost a perfect fluid, but when in motion it possesses the property of viscosity—that is, the friction among the particles of the fluid resists a change of form or distortion, but not a change in volume. A force producing such changes is proportional to the rate at which such changes take place.

The whole of the laws which are used for estimating the power required for the propelling machinery are based on experimental data, and only the frictional resistance of the under-water surface of the ship's sides (called the *wetted* surface) can be measured with any degree of accuracy. This measurement is obtained from model experiments or actual ships.

2. **Frictional Resistance.**—If a very thin plank is completely immersed and towed through the water, it is found that :

(*a*) Frictional resistance F increases with a greater co-efficient, μ, of roughness of the surface ; this is the resistance with which we have to deal in sea-going ships—

$$(F \propto \mu).$$

(*b*) Frictional resistance increases as about the *square*, actually the 1·84 power of the speed S or velocity—

$$(F \propto S^{1\cdot84}).$$

(*c*) Frictional resistance increases directly as the area A of the surface immersed—

$$(F \propto A).$$

Frictional resistance increases with increase of depth of immersion consequent on the greater pressure (or head) of water on the deeper surface ; for the surface type of ship the increase is of small importance.

Frictional resistance increases with the length of the immersed

surface, but the greater the length the *less* is the friction per unit of area, because the particles of water coming under the cleaving and dragging influence of the fore part of the body obtain a movement in the same direction as the body.

An increase of area A in similar proportion would increase the depth and thereby increase the friction per unit; an increase of A would also increase the length and thereby decrease the friction per unit; and therefore it may be taken as true that:

Frictional resistance increases directly as the area A of the immersed surface—

$$(F \propto A).$$

(*d*) Combining (*a*), (*b*), and (*c*)—

$$(F \propto A \,.\, \mu \,.\, S^{1{\cdot}84}).$$

If the plank has 100 square feet of clean varnished surface, such as is found on a newly docked and painted ship, the power required to move it through the water at a speed per hour of—

(10×1), or 10 knots, is from 4 to 5 horse-power, or (4×1^3);
(10×2), or 20 knots, is from 32 to 40 horse-power, or (4×2^3);
(10×3), or 30 knots, is from 108 to 135 horse-power, or (4×3^3);

or, in other words, the horse-power required to overcome the frictional resistance of a newly docked and painted ship varies as about the *cube of the speed.*

As a ship goes through the water the rough surface of the sides causes a dragging action, which induces the water to flow at some speed which is, of course, less than that of the ship. This induced flow forms the medium in which the propeller works, and thus the propeller acts on water which is already set in motion. The speed of the water thus set in motion by the frictional wake is found to be something below 10 per cent of the speed of the ship.

3. **Eddy-making.**—Eddies are caused by the varying velocities set up by the dragging action of the ship. The particles roll over each other, and, accumulating varying velocities and energy, react on each other. All this disturbing action is a waste of the propelling force. In the same manner—that is to say, both by lack of fairness of form and by surface friction—the action of the propeller blades sets up another series of eddies, due to the blades acting obliquely to the direction of motion and to the frictional drag of the surfaces of the blades.

In well-formed ships, at low speeds, the total eddy-making resistance is small—not exceeding 10 per cent of the total resistance, and is sometimes less. Eddying resistance is principally due to the defective form of entrance and run, particularly the latter, because for structural reasons the stern lines cannot usually be made quite as fine as those at the bow.

4. **Wave-making.**—A floating ship moving through a compressible medium such as air, or partly in a compressible and partly in an incompressible medium, tends to make a wave or waves.

If a vessel be of a perfect wave form and moves at a speed in knots equal to 6080 divided by 1·3 times her length in feet, she will make *one* wave.

If this vessel moves faster or slower than the above speed, she will make more than one wave—possibly three : one diverging at the bow, one transverse due to body, and one of replacement at the stern—according as her lines are then too fine or too full for the speed. In a vessel built on wave form, the three waves are more or less in evidence.

A ship with a comparatively low proportion of power to displacement, such as an ordinary sea-going vessel or sailing boat, starts by gradually increasing her waves, and eventually is unable to rise out of the wave due to her length, but a vessel with a large power can rise out of it.

A wave is due to the particles of water disturbed by the passage of the vessel trying to regain equilibrium, which they do by an oscillatory motion in a perfect fluid—but in water there is a certain actual motion.

A vessel going " slow " may give the displaced water so much time to regain equilibrium that it has too much leisure to make a wave, but a vessel going very fast may not allow time for a wave to be made of the size that would obtain if going " slow."

Flaring bows, or flat sterns, or other peculiarities in lines, may by pressing on wave crests reduce the size and utilise some of their energy by lightening the vessel's displacement for a time.

The speed of a wave is retarded by shallow water, and therefore vessels which have insufficient power to rise out of a wave also have their speed retarded. But a vessel whose power enables her to rise out of the wave can get near the crest positions sooner when the wave is slower, and the vessel can thus go faster in shallow water ; this has frequently been noticed in trials of torpedo craft.

For warships and other fast vessels with fine lines, wave-making is

comparatively unimportant at moderate speeds, but it absorbs a large proportion of the power, probably 15 to 20 per cent, at the highest speeds.

Augment of Resistance.—In screw-propelled ships, the ship or hull resistance is increased or augmented by the action of the screw in diminishing the pressure of the water under the counter or stern end of the ship; and at a certain high speed, which is peculiar to every individual vessel, any further increase of speed becomes impossible. In some cases the augment of resistance has been confounded with that caused by the defective form of screw employed, which actually limited the speed, and not the form of the hull to which resistance augmentation is properly due.

In the trials of the old *Medea* class an addition of over 30 per cent in I.H.P. produced a doubtful increase of about one-tenth of a knot in speed; subsequently in the *Latona* class, in all respects similar except an increased length of 35 feet and a consequent increased displacement of 800 tons, a higher speed was obtained without the additional 3000 I.H.P.

"Mauretania" Model Experiments.—Probably the most interesting and important experiments made and published in recent years were made by Messrs. Swan, Hunter, and Wigham-Richardson, and the Wallsend Slipway and Engineering Company, with a model of one-sixteenth the actual dimensions of the *Mauretania*, and about $47\frac{1}{2}$ feet in length. The trials were carried out in a dock on the Tyne, and events have proved their great accuracy in actual practice. The motive power of the model was supplied by electric motors driving four screws and shafts, for which power was supplied by accumulators stowed in the launch itself. Considerable difficulty was at first experienced by rapid fouling of the wetted surface, but eventually this element of inaccuracy was overcome. Incidentally it may be mentioned that, after seven weeks, the foulness of the bottom caused an increase of 25 per cent in power at full speed, and differences quite as large obtained after only two weeks when the fouling was not interrupted by speed trials. Dynamometers were carefully arranged for taking readings under various conditions, and the instruments were tested from time to time by means of a special form of Prony brake. Several hundreds of trial runs were made, and the actual pitch and diameter and other particulars of the screws obtained. There appeared to be no material difference by using either twin or quadruple screws, and eventually four screws were adopted, the outward pair being placed 78 ft. 11 in.

before the middle pair ; in this position the forward propellers did not appear to interfere with the efficiency of the after pair.

Influence of Depth of Water on Speed.—For many years past it has been known that shallow water affected the speed which might otherwise be obtained for the same vessel. The difference in resistance (it is now known that it may be either increased or decreased) is due to wave-making ; and the form of wave created is now generally termed the " canal wave." Investigations by Messrs. W. W. Marriner and H. E. Yarrow, by Messrs. Thornycroft, and by Colonel Giuseppe Rota of the Italian Navy, have been supplemented and summarised by Mr. Sydney Barnaby in the Watt Anniversary Lecture at Greenock, January 19, 1906. (See *Engineering*, February 2, 1906.)

These results are somewhat curious and appear to be applicable to all high-speed vessels. Roughly, for comparative purposes of various classes of vessel, the depth of water should be 10 to 11 times the load draught of the vessel, when of fine lines with a length of about 9 or 10 times the beam. In shallower water the speed gradually decreases for full power, until the depth is only $5\frac{1}{2}$ to 6 times the draught of water, which is about the most unfavourable condition for full power. With a further decrease of depth, the speed gradually increases until it becomes a maximum at a depth of about twice the load draught. But at half power the drag of the stern wave, due probably to its greater wave length, decreases the speed very much below that obtainable for the same power in deeper water.

With ships of fine lines, but of a length of only 6 to 7 times the beam, such as old battleships and cruisers, the problem becomes more complicated. To be on the safe side, it is probably in water of at least 11 times the draught that the best speed will be obtained at all powers. The *Mauretania* model experiments were run in a depth of 24 feet, the draught of the model being $24\frac{3}{4}$ inches.

[TABLE

INFLUENCE OF DEPTH OF WATER ON SPEED.

Depth of Water.	Power or Speed.	Remarks on Effects.
27-knot German Torpedo Boat, 200 ft. long, 347 tons displacement.		
Various	12 knots	Inappreciable
130 ft. and upwards	12 to 17 knots	Insignificant
23 ft.	Half power	5·64 knots *less* than in 200 ft. depth
23 ft.	Full power	0·84 knots *more* than in 200 ft. depth
80 ft.	27 knots	Most unfavourable
25½-knot Torpedo Boat Destroyers (River Class), 225 ft. long, 600 tons.		
25 to 35 ft.	Full power	Highest speed obtained
120 ft. and upwards	,,	Most favourable
50 ft.	,,	2 knots *less* than in 100 ft. depth
240 ft.	,,	1 knot *more* than in 100 ft. depth
60 to 70 ft.	,,	Most unfavourable
26-knot Torpedo Boats or Coastal Destroyers, 168 ft. long, 230 tons.		
20 ft.	Full power	Most favourable
33 ft. (?)	,,	Most unfavourable
98 ft. and upwards	,,	No appreciable increase
33-knot Sea-going Destroyers, 260 ft. long, 800 tons.		
32 ft.	Full power	Most favourable
50 ft. (?)	,,	Most unfavourable
150 ft. and upwards	,,	No appreciable increase
36-knot Sea-going Destroyer, 345 ft. long, 1800 tons.		
36 ft.	Full power	Most favourable
55 ft. (?)	,,	Most unfavourable
170 ft. and upwards	,,	No appreciable increase

Air Resistance.—The resistance of the atmosphere is becoming of increasing importance, because the surface exposed to the atmosphere and the speed are steadily increasing. In very fast vessels, careful design has probably reduced air resistance ; but the precautions taken by contractors to remove or minimise all possible obstructions show the importance attached to it. A destroyer steaming against the wind causes an indraught abaft the forecastle or other obstruction, similar to the windage of a railway train.

Mr. Froude's experiments showed that at a velocity of one foot-second the air resistance on a plane area is 0·0017 lb. per square foot. Assuming that the athwartship area of a 30-knot destroyer, which is exposed to the air, is about 200 square feet, the velocity 50 foot-seconds (or 29·5 knots), and that the air resistance varies as the square of the velocity ; then—

$$\text{Total air resistance, in pounds} = (200 \times 0{\cdot}0017) \times 50^2$$
$$= 2 \times 17 \times 25 = 850 \text{ lb.}$$

If instead of steaming through still air, as above, the destroyer is steaming against a 15-knot breeze, the resistance is increased to about 2000 lb., and the I.H.P. required to overcome this resistance is—

$$R \times S = \frac{2000 \times 75 \times 60}{33{,}000} = 275 \text{ nearly.}$$

About 6000 I.H.P. is required to propel this destroyer at 30 knots, and the percentage of this expended in overcoming the air resistance when steaming against a 15-knot breeze—

$$= \frac{275 \times 100}{6000} = 4{\cdot}6 \text{ per cent.}$$

In the *Mauretania* model experiments it was demonstrated that 12 per cent more power would be required to propel the vessel at 25 knots against a 25-knot wind than when moving through still air.

Influence of Weather.—Throughout these remarks it has been assumed that the vessel is propelled through still water. In practice, of course, there is a large difference, due to the roughness of the sea, produced by the wind alone, or wind and current. Sea speed is best maintained with great length, which generally involves increased displacement, and even when both these are incorporated the influence of form or shape may neutralise the advantages which might be otherwise expected. A 20 per cent increase in power, and a decrease of 20 per cent in speed, is not unusual when steaming against a head sea, and the difference is principally due to the pitching motion. If the ship can be turned about so as to take up a rolling motion when running before the sea, the speed may be increased, and the power decreased to normal. Experiments in smooth water, on which nearly all theoretical investigations are based, are not always suitable to sea-going purposes. Results should be tabulated of the experience gained under sea-going conditions, and these should be shown on a large-scale curve for further reference. At present there are very little sea-going data on which to base any reliable calculation for future designs, although such details can be obtained in everyday experience.

Effect of Sea-going Conditions on Power and Speed.—In the tabular statement below, the effect of various changes of steaming conditions is very fairly shown, but in some of the results the speed does not appear to have been very accurately logged. The revolutions

at very high speeds generally give a more nearly accurate measurement. The effect of greater depth of water is shown by a comparison of the contractors' trials off Chesil Beach (depth 17 to 20 fathoms) in September 1902, and the 8 hours' trial between Vigo and the Eddystone (depth 70 to 200 fathoms) on April 29, 1907. The conditions are almost identical except for the greater depth of water, which gave an increased speed of 0·4 knot.

The effect of increased frictional resistance of the wetted surface is shown by comparison of trials of September 24, 1902, and December 12, 1907; there is a slight gain in speed due to increase of depth of water, but nearly 2000 I.H.P. more are required to obtain the same number of revolutions. In February 1907, owing to foulness of wetted surface, the rate of revolution could not be increased above 121 per minute although the engines were supplied with all the steam which they could utilise; the speed was logged at only 23 knots, but the revolutions should have produced 24·2 knots with a clean bottom. Under almost identical conditions, but with a clean bottom, a speed of 24·5 knots was obtained in April 1907.

A careful study of this statement together with the coal consumption on actual service at moderate speeds, shows that the period elapsing between the docking of large battleships and cruisers should not generally exceed six months. Docking for purposes of cleaning should be considered in conjunction with the coal and water bills. Apart from considerations of pure economy, frequent docking is a necessity for war purposes; only a small number of docks is available, and on the brink of war it would be impossible to dock all the ships at once. A loss of 1 to $1\frac{1}{2}$ knots might also be attended with dire consequences.

An analysis of relative costs in power for moderate increases in speed in slow and high-speed vessels is given in an article in *International Marine Engineering* of May 1919, by Sydney Graves Koon, M.M.E., and refers particularly to the *Drake* and her sister vessels *Good Hope*, *Leviathan*, and *King Alfred*, of which the *Drake* and *Good Hope* were both lost during the war.

[TABLE

H.M.S. *DRAKE*

Date of Trial.	Displacement in Tons.	Direction and Force of Wind.	State of Sea.	Time since last Undocked.	Duration of Trial.	I.H.P. Average.	I.H.P. per 1 Ton of Coal per Hour.	Revolutions per Minute.	Logged Speed in Knots.
				Months.	Hours.				
June 1902 .	14,216	Various, 2	Moderate to slight	1	8	30,849	1193	116	23·05
Sept. 23, 1902.	14,158	Various, 1 to 2	Smooth	$\frac{1}{4}$	4 runs off Chesil Beach	31,454	...	122·4	24·11
June 1902 .	14,348	Various, 2	Moderate to slight	1	30	23,004	1240	105·4	22·08
Sept. 24, 1902.	14,100	Various, 1 to 2	Smooth	$\frac{1}{4}$	4 runs off Chesil Beach	22,534	...	111·5	22·16
The above were contractors' trials, with original and new propellers.									
June 29, 1903.	13,955	Ahead, 5	Slight swell	6	3	28,671	1125	119·23	23·0
Feb. 27, 1904 .	13,259	Beam	Head swell	$4\frac{1}{2}$	...	24,554	929	115·35	23·5
July 5, 1904 .	14·941	Bow, 1 to 2	Smooth	1	8	28,184	986	116·2	22·8
Jan. 5, 1905 .	14·941	Bow, 2	,,	2	8	29,690	1004	120·08	24·16
June 23, 1905.	14,187	Bow, 1	,,	1	8	29,201	953	119·615	23·72
Oct. 31, 1905 .	14,941	Bow, 3	,,	5	8	31,061	969	121·4	24·25
Mar. 1, 1906 .	14,100	Various, 2	,,	9	8	28,632	957	117·6	23·5
June 29, 1906.	15,144	Ahead, 2 to 5	Moderate	$3\frac{1}{2}$	7 Manœuvres	Unreliable	...	107·0	20·4
June 15, 1906.	14,738	Various, 2 to 3	Moderate to slight	$3\frac{1}{4}$	1 Manœuvres	32,200	949	123·0	24·8
Aug. 25, 1906.	14,448	Beam, 3 to 6	Rough	$5\frac{1}{2}$	8	31,338	1036	120·0	23·2
Dec. 12, 1906 .	14,883	Various, 2 to 6	Smooth to heavy ocean swell	9	30	24,459	990	111·3	22·3
Feb. 1907 .	14,421	Various, 2 to 3	Slight swell	$11\frac{1}{2}$	$1\frac{1}{2}$ Manœuvres	30,006	1058	121·0	23·0
April 29, 1907.	14,593	Bow, 1	Smooth	$1\frac{1}{2}$	8	31,717	1016	122·5	24·5

Power and Displacement.—The relation between these two quantities is expressed by the Admiralty formula :

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}} \times S^3}{C}$$

in which D = Displacement in tons.
S = Speed in knots, and
C = a constant, with values of about 200 to 400 according to speed and class of vessel.

Another formula devised by Captain H. C. Anstey agrees very closely with practice when the speed in knots divided by the square root of the length in feet is greater than unity.

$$\text{I.H.P.} = \frac{S^3 \times \sqrt{D}}{50 \text{ to } 55}$$

$$\text{S.H.P.} = \frac{S^3 \times \sqrt{D}}{60 \text{ to } 65}$$

$$= \frac{S^3 \times \sqrt{D}}{C}$$

in which C = about 62 for old types of warships, and
= about 65 for pinnaces and launches.

The ratio between S.H.P. and I.H.P. admits wide differences of comparison. I.H.P. is used for reciprocating engines and S.H.P. for turbines, and the difference is entirely dependent on the *mechanical* efficiencies of the two types of engine.

Curves of Power and Speed.—By combining the speed and power, a curve can be constructed to show the power required for any speed within the limits of the curve, and for a short distance above and below these limits.

In Fig. 361 the horizontal line shows the speed in knots, and the vertical lines the powers required for the various speeds of recent warships. The curve is drawn through the various points obtained from results of trials. By very carefully plotting, on a large-scale curve, the lower powers and speeds, and then continuing the curve until it cuts the vertical zero ordinate, it is found that some power is apparently indicated (as shown by the curve) when no motion is imparted ; the power thus registered is that required to overcome the frictional resistance of the engines.

Power and Speed.—Within the ordinary limits of power (between one-fifth and three-fourths that of full power) and speed, only the frictional resistance need be considered. It varies as about the 1·84

power of the speed as found by experiment; but for ordinary purposes the *square* is, perhaps, a closer approximation.

First law: *The resistance varies as the square of speed* ($R \propto S^2$).

Let S = speed in feet per minute, and let R = resistance in pounds at this speed. (R is the actual pull on a tow-rope required to move the vessel at a speed S.)

$$\text{Then } R = K \times S^2$$

where K is a constant.

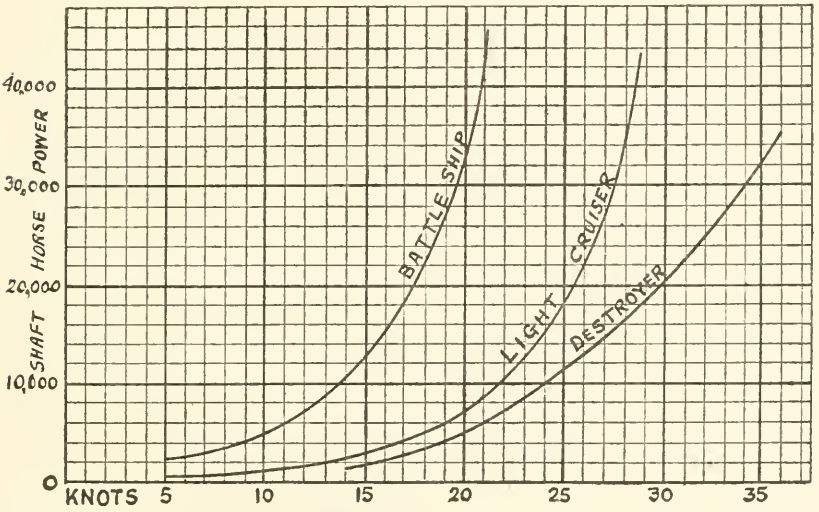


FIG. 361.—Curves of Power and Speed.

Multiply both sides of this equation by S; then—

$$R \times S = K \times S^3.$$

But $R \times S$ = work done in overcoming a resistance R through a space S, and, if expressed in foot-pounds and divided by 33,000, is equal to the I.H.P. or S.H.P. required to overcome the resistance.

Second law: *The S.H.P. varies as the cube of the speed* (I.H.P. or S.H.P. $\propto S^3$).

Fuel, Speed, and Power.—Within moderate limits the consumption of fuel per hour per I.H.P. is the same at any power, and—

$$\text{Consumption of fuel per hour} \propto \frac{\text{S.H.P.}}{\text{speed}} \propto \frac{S^3}{\text{speed}} \propto S^2.$$

Speed in knots (per hour)

Therefore—

Third law: *The consumption of fuel* (*F*) *per mile varies as the square of the speed*, or $F \propto S^2$.

EXAMPLE.—If 40 tons of fuel are burnt at a speed of 20 knots, how much would be burnt at 21 knots?

Evidently the consumption (f) per mile = 2 tons, and

$$\frac{F}{f} = \frac{S^2}{s^2} = \frac{441}{400}.$$

Therefore $$F = \frac{S^2}{s^2} \times f = 2{\cdot}2 \text{ tons per mile}$$

$$= 46{\cdot}2 \text{ tons per hour.}$$

Fourth law: *The total fuel consumption* (*F*) *for any distance* (*D*) *varies as the square of the speed multiplied by the distance*, or $F.D \propto S^2D$. This is evidently the same as the third law, each side being multiplied by the distance D, and it assumes that the fuel consumption per horse power is constant at all speeds which is not accurate (see Fig. 362).

EXAMPLE.—If 200 tons of coal are consumed in steaming 800 nautical miles at 11 knots, how much will be required for a distance of 1210 nautical miles at 10 knots—excluding the coal used for auxiliary purposes in each case?

Total consumption $$F.D = 200 \text{ tons}$$

and $$\frac{f.d}{F.D} = \frac{s^2.d}{S^2.D}.$$

From which $$f.d = 200 \times \frac{100 \times 1210}{121 \times 800} = 250 \text{ tons.}$$

The steam trials of the Japanese battleship *Kashima* are a fairly recent demonstration of the value of n in the theoretical equation, $I.H.P. \propto S^n$, which show that at about full power the $I.H.P. \propto S^{3{\cdot}16}$, and at about one-third power the I.H.P. varies as $S^{2{\cdot}87}$, both values being based on the correctness of the observations at a power of about one-sixth the full power.

H.I.J.M.S. *KASHIMA*

Revs. per Min.	I.H.P.	Speed in Knots.	Value of n, the Index of S^n.
69·5	3,030	11·136	..
89	6,275	14·27	2·87
102	9,160	16·323	2·87
110	11,400	17·204	3·02
113·6	13,000	18·000	3·01
123	17,280	19·242	3·16

Comparisons of several naval vessels, of different types, show the same practical tendency, but unfortunately the general inaccuracy of the speed, *by log*, at one-fifth power renders in this connection nearly the whole of these comparisons valueless. As an example, the *New Zealand* at 16·9 and 18·6 knots has index values of 1·9 and 2·11 respectively; but if a comparison is made between these two speeds and the respective powers, independently of the logged speed of 9 knots (obviously inaccurate) at one-fifth power, the index value is 3·6, which is probably more nearly correct.

Willans Line.—From some investigations by Mr. Willans on fuel consumption of small engines it appeared that the ratio between fuel

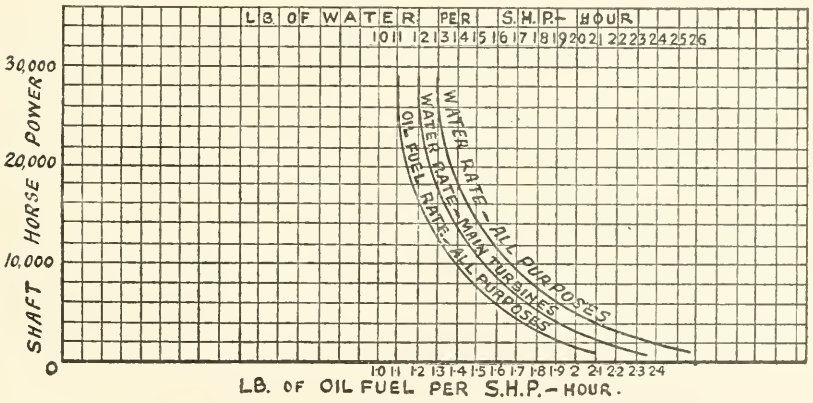

LB. OF WATER PER S.H.P. HOUR
10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26
30,000
20,000
10,000
0
SHAFT HORSE POWER
WATER RATE – ALL PURPOSES
WATER RATE – MAIN TURBINES
OIL FUEL RATE – ALL PURPOSES
10 11 12 13 14 15 16 17 18 19 2 21 22 23 24
LB. OF OIL FUEL PER S.H.P. – HOUR.

FIG. 362.—Steam and Fuel Consumption.

consumption and power was very nearly constant, and therefore the curve would be almost a *straight-line* curve. With large engines of great power this is not the case as will be seen from Fig. 362.

Fuel and Steam Consumption per S.H.P. Hour.—Curves are shown in Fig. 362 of the water (steam) and oil fuel consumptions of a recently constructed small displacement but high powered vessel. From this it appears that, within the limits of the curve, the steam produced is in exact ratio of 10 to 1 of the fuel burnt, which is not improbable when burning oil fuel. There is a steady and gradual increase in consumption of both steam and fuel as the power *decreases*. It will also be noticed that the steam consumption for all purposes is considerably greater than that for the main turbine propelling engines only, with a steadily increasing ratio at the low powers, at which the auxiliary

consumption assumes a very large proportion of the total consumption. This affects very considerably the distance run per ton of fuel, and is very clearly illustrated in the curve shown in Fig. 363, where the distance run of $13\frac{1}{2}$ nautical miles at a speed of 13 knots steadily decreases to less than $2\frac{1}{2}$ miles at a speed of 35 knots.

In all estimates of the distance run, or economical speed, the effect of the auxiliary consumption must be allowed for.

Economical Speed.—The greatest distance which a ship can steam on a consumption of 1 ton of fuel is the most economical speed ; but for any vessel this distance is practically indeterminate because of the variation of wind and sea conditions, the cleanliness or otherwise of

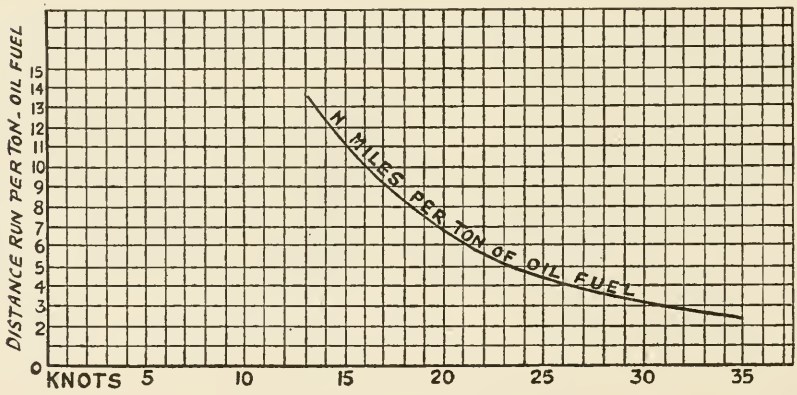


Fig. 363.—Distance Run per Ton of Fuel.

the ship's bottom, the quality of the coal, the consumption of fuel and steam for auxiliary purposes, and other minor reasons. Generally for large ships whose I.H.P. is nearly equal to the tonnage displacement, the economical speed is at about one-fifth to one-sixth power. In practice it is necessary to construct a simple tabular statement—including in it the fuel used for auxiliary purposes—of the fuel consumption at each knot speed between the lowest obtainable and the full speed, and to calculate the distance steamed per ton of fuel at each of these speeds.

A rough estimate of the loss from the state of the ship's wetted surface is 1 per cent of the full speed for each month since the ship was last undocked ; this is fairly accurate up to a period of six months, but after this the loss is only very slightly increased.

In Fig. 364, curves have been drawn to show *BB* the coal burnt per hour in tons ; the coal consumption *CC* per I.H.P. per hour in lb. ; and the distance run *AA* per ton of coal burnt relative to the speed in knots. The coal used for auxiliary purposes is included in the consumption per hour in tons and in the distance run, but not in the consumption per I.H.P.-hour. From the curve it appears that the economical speed is sharply defined at E, for 10 knots ; but this is only the case when the conditions are identical and in exact accordance with the conditions thus represented.

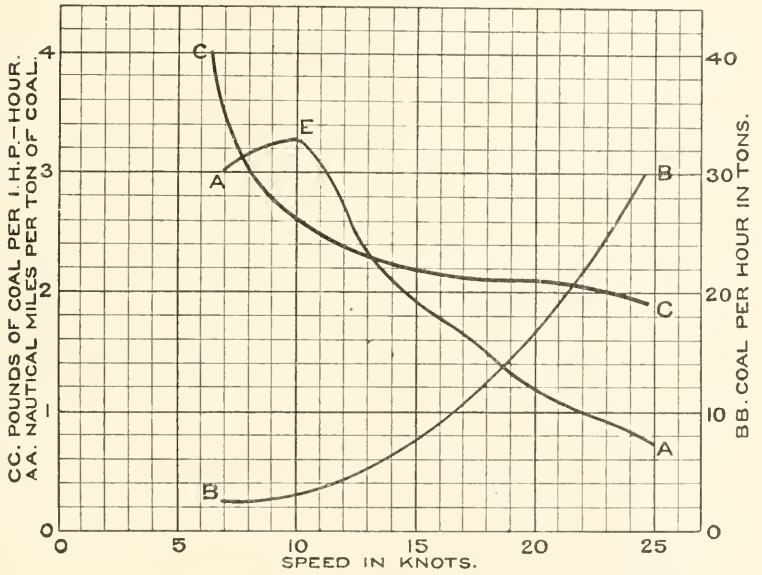


Fig. 364.—Curve of Performance.

Taking all the above remarks into consideration, it may be stated that the economical speed for any vessel is probably about two-fifths of the full speed, obtainable with full power, corresponding to the state of the wetted surface. This speed under ordinary conditions in recently constructed vessels would correspond to a power somewhere between one-fourteenth and one-twelfth the full S.H.P.

Propulsion.—A vessel is caused to move through the water and over its surface by the propellers moving the water in such a way that motion of the vessel is produced in the required direction. The resistance must be overcome to produce motion, and the displacement of

the water is a measure of the resistance. "Action and reaction are equal and opposite," and if all the resisting forces produced by the effort of propulsion be resolved into two parts, one of which is in line with the direction of motion of the vessel, and the other at any angle with this line, then the first may be considered as a measure of the work usefully expended, and the remainder as work expended in creating wasteful disturbance. From which it may be considered that the effective thrust is equal to the momentum of the stream driven directly astern.

For paddle-wheel propulsion, the area of the stream propelled sternward is evidently equal to the area of the columns acted on by the floats on each side of the ship ; and if the floats are totally immersed it is approximately equal to the area of two floats, one on each side of the vessel.

For screw propulsion, the area of the stream is approximately equal to the area of a circle, whose diameter is that of the screw *less* an area due to the greatest area of the boss of the screw, which has no propelling effect.

Effective Horse Power and Thrust.—The I.H.P. is a measure of the work imparted to the pistons, and part of it is used in overcoming the frictional resistances of the mechanism of the engine itself, the thrust mechanism and transmission gearing (if fitted), and the propeller, and also in wasteful disturbances caused by the oblique and other action of the propeller.

The S.H.P. is a measure of the useful work imparted to the shaft, and part of it is used in overcoming the resistance in the thrust and transmission mechanism as well as the propeller resistances and waste as above stated.

The Effective thrust, or the proportion of the total thrust of the propeller, or propellers, which is effective in thrusting the ship through the water, is thus promoted by only a moderate proportion of the I.H.P. or S.H.P. This proportion, which may amount to between 50 and 70 per cent, is called the Effective or Thrust Horse Power, T.H.P. or E.H.P.

Measurement of Effective Thrust.—Messrs. Cammell-Laird have introduced a very interesting and valuable contribution to science and practice in a form of apparatus connected with the Michell thrust block system.

Fig. 365 shows the general arrangement as fitted in H.M.S. *Mackay* and Fig. 366 the results as plotted on curves of thrust, power, and speed.

Behind each of the pivots of the Michell segments, a small cylinder is fitted which is supplied by hand or otherwise by oil pressure just sufficient to force the abutment ring off its metallic seat, and thus suffices to bear the whole of the propeller thrust. A total fore-and-aft movement of $\frac{1}{16}$ of an inch is allowed for this purpose. By calculation, knowing the area and number of the rams (fitted in the cylinders), from the gauge pressure applied as oil pressure to the rams, the total

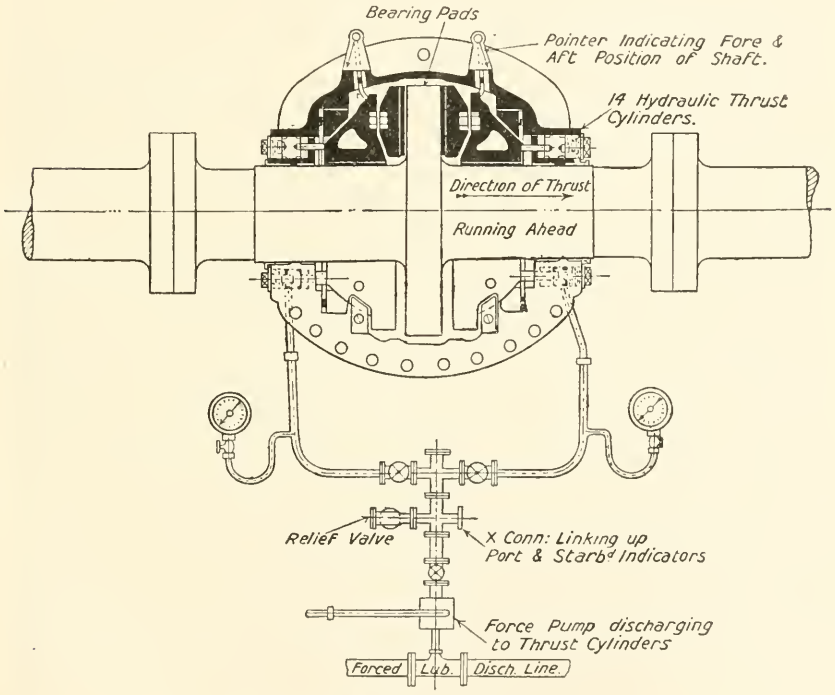

Bearing Pads
Pointer Indicating Fore & Aft Position of Shaft.
14 Hydraulic Thrust Cylinders.
Direction of Thrust
Running Ahead
Relief Valve
X Conn: Linking up Port & Starbd Indicators
Force Pump discharging to Thrust Cylinders
Forced Lub. Disch. Line.

FIG. 365.—Measurement of Propeller Thrust.

thrust can be ascertained at any instant and therefore for any variation in speed and power.

On the trials of the *Mackay* the loads registered were 59 and 56 tons respectively for the port and starboard shafts, and these loads varied at the end of measured mile, when turning, to 61 and 47 tons, reverting to the former figures when on the straight course again. The chart clearly shows the beginning of cavitation where the curve falls away at about 430 revolutions per minute. (*The Engineer*, Aug. 29, 1919.)

For purposes of illustration of principle in application the following will serve as an example.

From the curves in Fig. 366 :

Under heavy displacement at 30 knots, the S.H.P. is 33,000, thrust 98 tons, and revolutions per minute 305 ;

Under light displacement at 30 knots, the S.H.P. is 26,000, thrust 70 tons, and revolutions per minute 286.

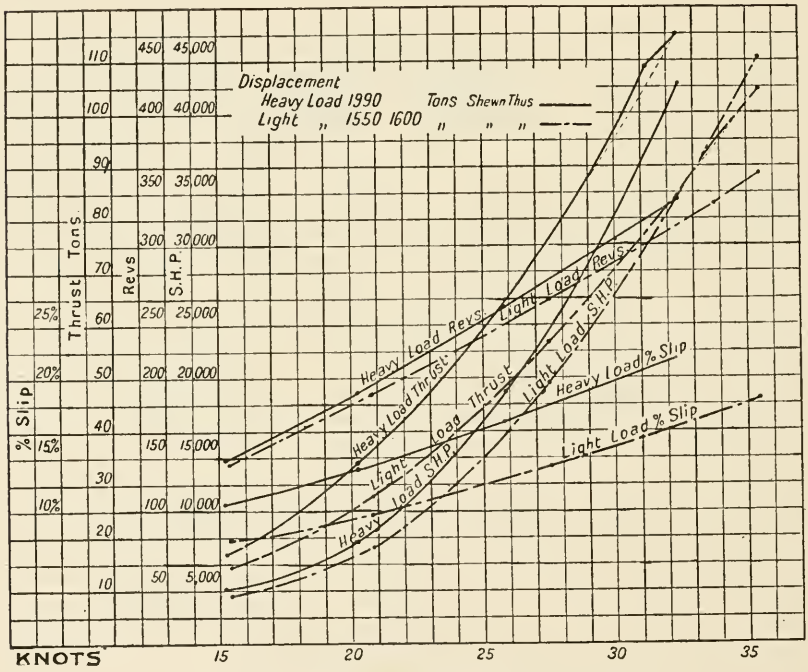


Fig. 366.—Curves of Propeller Thrust, Slip, Power of Speed.

The apparent slip under heavy load is about 19½ per cent, and under light load it is 15 per cent ; by using these as a check on one another, observing that the speed is the same in each case, it can be calculated that the pitch of the propellers is about 12½ feet, and consequently the distance through which the resistance is overcome per minute (= Pitch × Revolutions per minute) is

under heavy load = $305 \times 12\frac{1}{2}$ feet per minute, and
under light load = $286 \times 12\frac{1}{2}$ feet per minute.

The I.H.P. is not available and the estimate of the thrust efficiency, E, must be given as a fraction or percentage of the S.H.P.

The equation is :

$$\text{Thrust in lb.} = \frac{\text{foot-lb. of work done per minute}}{\text{distance through which resistance is overcome per minute.}}$$

Under heavy load :

$$98 \times 2240 = \frac{33{,}000 \times 33{,}000}{305 \times 12\frac{1}{2}} \times \text{E.}$$

From which, E = 77 per cent nearly.

Under light load :

$$70 \times 2240 = \frac{26{,}000 \times 33{,}000}{286 \times 12\frac{1}{2}} \times \text{E.}$$

From which, E = 65·3 per cent about.

Assuming that in this instance single reduction gearing is fitted and that the S.H.P. is measured between the turbines and the gearing, then from the two efficiencies calculated above must be deducted some allowance for losses in the gearing transmissions and in the shaft bearings abaft them ; a fair amount would be 4 per cent deduction. and thus the propeller efficiencies would become :

Under heavy load . . 96% of 77% = 74% nearly.
Under light load . . 96% of 65·3% = 62·7% about.

But if the S.H.P. is measured abaft the gearing, which is probably the case as the position is more convenient for the measurement, then the deduction is merely a nominal one of about 1 per cent for the loss in the shaft bearings, and thus the propeller efficiencies would become :

Under heavy load . . 99% of 77% = 76% about.
Under light load . . 99% of 65·3% = 64·6% about.

These efficiencies are high, but under the conditions assumed they refer particularly to the efficiencies of the *propellers*, the thrust of which is independent of the actual rate of movement of the ship. If the area of the propellers is considered, the actual thrust pressure exerted on each unit of projected or developed area can be calculated, and from this it will be known at what particular pressure cavitation begins under the heavy load as shown by the top end of the heavy load thrust curve.

For practical purposes in any vessel moving freely through the

water and under no constraint of towage or other cause, the efficiency of *propulsion* as an actual result in speed of the vessel is possibly of greater importance ; in this case instead of taking the actual revolutions of the propellers as the basis of distance travelled, the speed of the ship must be taken.

This can be taken from the curves in Fig. 366 for this particular case. Under heavy load, at 32 knots, the thrust is about 112 tons and S.H.P. 40,000 ; from which the calculated efficiency of *propulsion* is 67·5 per cent nearly. Under light load, at 35 knots, the thrust is 102 tons and S.H.P. about 41,000 ; from which the calculated efficiency of propulsion is 67·7 per cent nearly.

The two results, which differ very little under heavy and light loads, may be regarded as very satisfactory because they are high, due almost entirely to the reduction gear being well proportioned, and the propulsive efficiency is practically the same under both heavy and light load, which is an indication of good design.

Until this adaptation of the Michell thrust to the measurement of the Propeller thrust was introduced, there was no exact system of measurement except in the experimental tank with models of actual hulls proposed and gradually fined down to approximate to the lines of least resistance to propulsion, which, as pointed out above, is not exactly the same thing as resistance overcome by the propellers in relation to the thrust pressure overcome by them in urging the ship along or when tied up to a wharf or a tow.

Pull on a Tow Rope.—If a ship be towed through the water the effective thrust is *slightly less than* the pull on the tow rope ; *and* it is not an exact measure of the pull, because it depends on the various efficiencies involved. For ships of ordinary form and shape the tow-rope resistance is generally equal to about 90 per cent of the effective thrust.

For towing purposes, the propeller should be proportioned to work effectively and without undue slip, and a large blade area is necessary to take the thrust. Apart from the important consideration of tonnage displacement, the most effective towing ship is one with a propeller of large pitch and blade area. For this reason, as well as the small area of the thrust collar surface, *ships propelled by direct-driven turbines are not suited to towing purposes* ; but with thrust blocks properly designed for the purpose, turbine-propelled vessels might be made fairly effective towing ships at moderately low speeds.

In any vessel not primarily constructed for towing purposes, only

one-third of the I.H.P. of the towing ship is available for towing another ship of about the same tonnage displacement because, roughly considered, of the total I.H.P. developed by the towing ship, one-third is used in overcoming the internal resistances, one-third is used in overcoming the resistance of the towing ship, and the remaining one-third only is used in overcoming the tow-rope resistance, and therefore the maximum pull

$$= \tfrac{1}{3} \times \frac{\text{I.H.P.} \times 33{,}000}{\text{N} \times \text{pitch}} \times \tfrac{1}{2} \text{ about};$$

assuming the effective pull to be 90 per cent of the propeller efficiency of 55 per cent.

EXAMPLE.—For vessels of 18,000 I.H.P., pitch of screws about $16\frac{1}{2}$ feet, and N = 75 r.p.m. at about one-third power—

$$\text{Pull on tow-rope} = \tfrac{1}{3} \times \frac{18{,}000 \times 33{,}000}{75 \times 16\frac{1}{2}} \times \tfrac{1}{2}$$

$$= 80{,}000 \text{ lb.}$$

$$= 35{\cdot}7 \text{ tons about.}$$

CHAPTER XXVIII

THE PROPELLER—PADDLE, JET, SCREW, AND TURBINE SCREW

Paddle Wheels.—In the early steamships paddles were used for propelling the ships through the water, and under good conditions of wind, weather, and proper immersion they still give a fairly high efficiency in comparison with the screw propeller.

The *radial paddle wheel* is the simplest to construct, as the floats are secured directly to the arms radiating from the centre of the wheel. In this case the only time when the float acts perpendicularly to the water, the surface of which is always considered horizontal, is when the radius arm is vertically below the centre ; but at all other times the thrust is oblique, and consequently less efficient.

The *feathering paddle wheel* was introduced to overcome the disadvantages arising from this oblique action. A diagram is shown, in Fig. 367, of the usual method of feathering. A is the centre of the wheel, which is generally mounted on the projecting ends of the crank shaft D. B is another centre, horizontally in front of the shaft centre, and fixed to the outboard part E of the paddle box F, but inside the box itself. BK is a radius bar, and KC is connected with it at K by a pin-joint. KC is a lever, with its fulcrum at L, which is a point on the wheel circle ; usually L very nearly coincides with C. The shaft revolves the wheel, which carries the floats with it through the concentric medium of the pin-joints L. The radius arm BK is constrained to work in a circle round B as centre, and at the same time L is constrained to work in a circle round A as centre. These two constraints produce an alteration in the movement of the floats (as shown in the figure), and at their entry to and exit from the water they are more nearly vertical, thus producing a more direct thrust.

The disadvantages of the paddle wheel compared with the screw are—

1. Greater variation of effect when the ship is rolling or pitching ;

the floats are sometimes immersed too much and at other times too little.

2. The draught of water must be accurately maintained, or the floats will, if the ship be sufficiently weighted, be immersed to such an extent that propulsion will be difficult or inefficient. If weight be taken out, the floats will not get sufficient hold of the water. On long voyages, the coal burnt during the passage alters the draught of water and the immersion of the floats considerably.

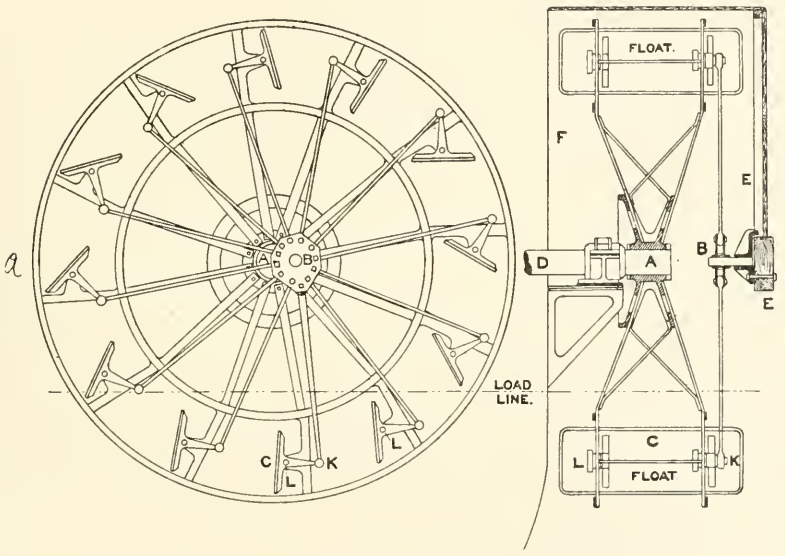


FIG. 367.—Feathering Paddle Wheel.

3. The paddle wheel is always exposed to shot and shell, and it is therefore of comparatively little use for warships.

Jet Propulsion.—Various attempts have been made to propel ships by pumping water through pipes. The water is drawn from the sea and ejected with a greater velocity, and the momentum produced by this acceleration constitutes the impelling force. The frictional resistances of the channels through which the water passes absorb a large share of the power, and in addition the changes in direction considerably reduce the momentum of the ejected water.

Two similar boats, built by Messrs. Thornycroft many years ago, one propelled by a screw and the other by jet propulsion, with a horse-power of about 170 each, obtained $17\frac{1}{4}$ knots and $12\frac{1}{2}$ knots respectively;

in other words, the screw-propelled boat could attain about the *same* speed for about *half* the power.

The screw in some ships, where partial tunnels are adopted to give a better run of the water to the propeller, is to some extent an extreme example of a jet propeller. If the tunnels are complete, as in the Thornycroft turbine arrangement used for shallow-draught boats (Fig. 372), the propeller simply performs the function of a helical pump with as little change in direction of flow of the water as possible.

Screw Propeller.—This is the usual method employed for propelling sea-going ships. The screw is placed under the counter or after-part of the ship, and is generally well immersed so that it is little affected by the disadvantages which render the paddle inefficient; and, in addition, the propelling engines can be kept below the water-line and, therefore, for warships, under a fair protection from shot and shell.

The screw consists of a boss, fixed on the tail end of a fore-and-aft shaft revolved by the engine, which carries a set of blades radiating from it, and these, being placed obliquely, form parts of the threads of an actual screw. When revolved, the screw advances through the water just as an ordinary screw advances through wood.

If an ordinary screw, such as is used by carpenters, be driven into a board by a turn-screw, the distance it advances into the wood in one complete revolution is one *pitch*. The marine propeller is simply a very short length of an actual screw (Fig. 368), but of very coarse pitch, and the thread of which is cut very deep, equal to about two-thirds the radius of the outside diameter. The ordinary screw will produce only one blade, as there is only one *continuous* thread; but if there are two threads, which lie parallel, side by side, and a short length be cut off, there are two blades, thus producing a two-bladed screw. Similarly, if there are three threads, there are three blades; and if four threads, four blades, and so on. Propellers generally have two or more blades, and this means that the screw is two or more threaded.

The *pitch* of the screw is the distance which the screw would advance in one complete revolution if it worked in a solid medium, such as steel or wood.

The *diameter* of the screw is the diameter of the circle described by the extreme point of the blades when the screw revolves in a fixed plane.

The *length* of the screw is measured along the axis, and is equal to the length of the blade measured parallel to the axis.

The *angle* of the screw is the angle contained between the blade and a plane perpendicular to the axis. It increases as the radius at which it is measured decreases.

The *disc area* is the area of a circle of which the diameter is equal to the diameter of the screw. More correctly, the disc area is less than the area, as defined above, by the area of the boss of the propeller at its largest diameter in a plane perpendicular to the axis.

The after side of the blade is called the *driving* face, while the forward side is called the *back* of the blade. The forward edge of the blade is called the *leading* edge, and the after edge is called the *following* edge.

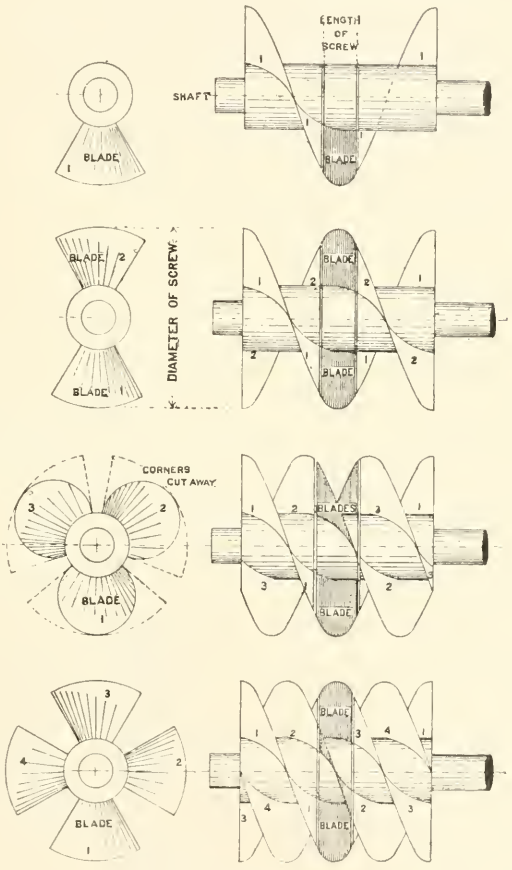


Fig. 368.—Screws and Blades.

The *developed area* of the blade is that measured on a plane parallel to the blade. The *projected* area is measured on a plane perpendicular to the axis, and is the developed area as projected on to this plane.

The *pitch-ratio* is the pitch divided by the diameter of the screw. When the pitch-ratio is great, the ratio of the projected area to the developed area is small.

Description of Screw.—Fig. 369 shows a longitudinal section

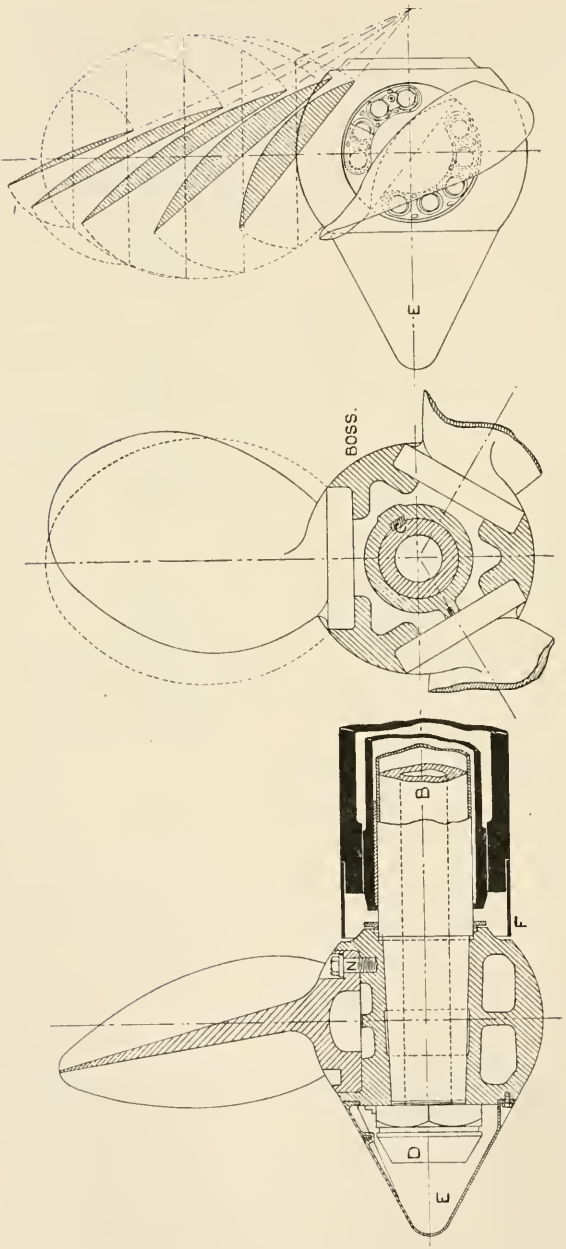


Fig. 369.—Screw Propeller.

through the boss and fittings of a large three-bladed screw; a section across the boss; and a plan of one blade with the bolt-holes and two bolts in place, and various cross sections of another blade, as used in some recent vessels.

For direct-drive turbine engines, the boss and blades are usually made in one piece of manganese bronze; but for reciprocating engines, the blades are generally made separately to the boss of the screw, and also for geared turbine vessels. Manganese bronze is generally used for the blades, when made separately, because with this material they can be made thinner and thus offer less cutting resistance; but steel or cast iron is sometimes used in trawlers and small slow vessels when the blades and boss are made in one piece.

The boss of the screw is secured to the shaft *B* by a steel feather *C* and a cap-nut *D*. The shaft is slightly coned, and the boss is driven or forced on and made a tight fit on the shaft. The cap-nut is prevented from slacking back by a guard plate, which is secured in the usual way by nuts and split pins on square-necked studs. At the forward end of the boss a packing gland is fitted, and generally consists of a flat ring, made in semicircular parts, with overlapping ends, pressed on to an india-rubber packing, slightly protruding from the recess into which it is jambed. The cap-nut and packing gland are fitted to prevent access of moisture to the shaft, between it and the boss, and from consequent corrosion.

A coned tail-piece *E* is fitted on the after end of the boss to cover the end of the shaft and nut, and to allow the water after passing between the propeller blades to take up a direct sternward and undisturbed flow without an eddying or cavitating wake. A guard *F* is fitted at the after end of the shaft bracket, and is shaped so as to allow a smooth run of water to the propeller; *F* protects the gland at the fore end of the boss, and also a guard ring, which keeps the lignum vitæ strips in place within the metal casing fitted inside the shaft bracket for carrying the shaft. The gun-metal casing fixed to the shaft to form its bearing surface extends to just abaft the india-rubber packing. The guard ring *F* is generally made in two parts, for convenience of removal, and sometimes of zinc, to protect other parts near it from corrosion. The gun-metal casing containing the strips is also made in two parts, and is generally designed to allow withdrawal from the fore end of the shaft bracket, without disturbing the propeller or shaft. As mentioned in Chapter XVIII., this casing must be removed before removing the shaft.

Variation of Pitch.—The blades of the propeller are flanged circularly near the root, and this flange fits into a space on the boss, accurately turned and fitted for the purpose. Gun-metal or bronze screws *N* secure each blade to the boss; the holes are elongated in the flange for the screws, and thus the blade can be set at a slightly different angle, with a corresponding difference of pitch of the screw propeller. This variation is convenient for slight alterations for trial, and also when the boiler pressure is necessarily reduced. This reduction of pressure is not common in modern practice; but, if decided on, the reduction of pitch allows an increase in the number of revolutions, and

nearly the same power can be developed, but with less economy, as with the original pitch and revolutions.

After the blade has been temporarily secured and tested for correctness of pitch, filling pieces are fitted in the recesses on each side of each securing screw; these are generally made of gun metal or lignum vitæ. Guard plates are fitted so that the screws cannot slack back, and there is generally one plate between each pair of boltheads. The plate is kept in place by the usual square-necked stud, through an elongated hole, nut, and split pin. A thin gun-metal covering plate is fitted over the boltheads and secured to the boss by countersunk screws. This plate is fashioned to present a fair surface to the passing water, and when closed up the boss of the propeller becomes almost globular in shape, except where the fore and after ends are flattened.

In the later and generally better shaped propellers great trouble is taken to secure—

1. A free and smooth run of water to the propeller and away from it, as it is important that the screw should act in water as little disturbed as possible.

2. A safe guarding of the securing arrangements of the propeller, its fittings, and surroundings. The proper inspection of these locking and securing arrangements is most important when a ship is docked.

Slip of the Screw.—The screw propeller does not work in a solid medium, but in water which yields slightly, and has also been set in motion by the passage of the ship. The difference between the distance that the screw would advance if it worked in an unyielding medium and the distance that it is actually carried by the ship in the same time is called the *apparent slip* of the screw, and is that usually shown in tabulated results.

The curves shown in Fig. 370 indicate the apparent slip of the screws in a fast destroyer, and it will be noticed that near full power when the destroyer makes an actual speed of 34 knots, the speed measured by the revolutions of the screw is nearly 46 knots: the difference being due to the apparent slip.

If p is the pitch of the screw in feet, and N is the number of revolutions of the engine *per minute*, then $p \times N$ is the distance the screw would have travelled in a solid medium in one minute; $(p \,.\, N)$ is called the speed of the screw.

If S is the speed of the ship in *feet per minute*, then—

The apparent slip $= p \,.\, N - S$; or

The apparent slip $= \frac{p \cdot N - S}{p \cdot N}$, expressed as a *fraction* of the speed of the screw; or

$$= \frac{p \cdot N - S}{p \cdot N} \times 100,$$ expressed as a *percentage* of the speed of the screw.

The *actual slip* is very difficult to obtain, because the screw is always acting in water already set in motion by its own action and the passage of the ship. It is evident that it must always be a *positive* quantity, or no thrust could be imparted; but in a few instances a *negative apparent slip* has been observed.

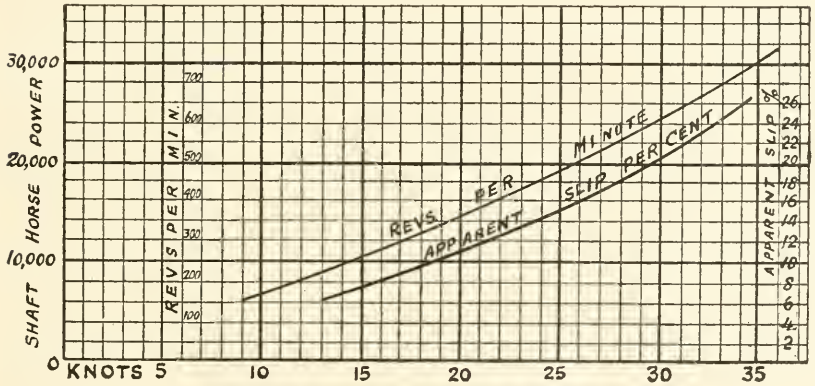


FIG. 370.—Curve of Apparent Slip per cent.

Negative Slip.—If the speed of the ship exceeds the speed of the propeller, the slip is said to be negative. In the equation quoted above, if the result is a negative quantity, the apparent slip is negative; and this is popularly known as negative slip, but is correctly negative *apparent* slip.

Negative apparent slip is obtained by the screw acting on the water set in motion by the ship, and following the direction of motion of the ship. With all propelled ships this following water, or following wake, is a usual occurrence. With ships with fine lines and suitable propellers, the following wake is not a serious loss, because the motion is imparted to a comparatively small column of water; but when the motion is imparted to a very large wake there is a great loss of propulsive effect, and although some of the energy imparted to the water

is returned in assisting the propulsion of the ship, a much larger amount is lost. It is particularly when this loss is great that negative apparent slip is obtained, and generally it may be stated that this attainment must result in an actual loss of economy and efficiency.

From experimental data obtained from ships of very fine lines and high speeds, it is found that a constant, or slightly increasing apparent positive slip, varying from 15 to 20 per cent, generally gives good results; and that the highest speeds are obtained with less power than with propellers which at the lower speeds showed an apparent positive slip of 5 to 10 per cent.

Direction of Rotation of Screws.—For vessels with a single screw the engine generally revolves with a right-hand motion when going ahead, or in the same direction as the hands of a clock when looking towards it; the screw is then right-handed, and the tops of the blade move towards the right when looked at from aft. With outward turning *twin* screws the top ends of the blades move away from the ship's hull when going ahead, but with inward turning they move towards the hull. Inward turning, the port screw then being right-handed and the starboard left-handed, came into fashion for a few years (1895 to 1901); but outward turning has been reverted to. There is no proved advantage in efficiency of propulsion with either system.

Comparative difficulties in manœuvring vessels are not due to the fact of screws turning either inwards or outwards, but rather to the relative position of the twin shafting. The shafts *diverge* from aft to forward with outward turning screws in many ships, whilst they *converge* with inward screws in others. Vessels, when turning, pivot about some point P (Fig. 371) in the keel line which oscillates from somewhere near the bow to about two-fifths the length of the vessel abaft it. The difference in manœuvring is caused by a mechanical turning moment $T \times Pb$ exerted along the respective shaft axes about the pivotal point. The outward turning screws being a little closer together and the engines farther apart, the force exerted by the thrust T is conveyed to a long arm Pb, while with inward turning the thrust is conveyed to a short arm; this difference naturally results in better manœuvring power with parallel or divergent shafting. It is even possible with convergent shafting to pull the head of the ship to port when going astern with the starboard screw, the turning moment being then negative, with the pivotal point well forward. The above contention is in agreement with everyday practice, and the mere

changing over of the propellers, port with starboard, would have no material effect on the handiness of any vessel; the difference lies entirely in the lines of shafting.

To obtain the minimum vibration, *single screw* vessels should always be fitted with either two or four blades. An uneven number, such as three, causes excessive vibration from the action on water divided into two streams by the stern post.

Twin Screws.—The diameter of the screw is limited by the draught of water and the greatest immersed transverse section of the ship. For ordinary sea-going purposes the tips of the blades should be immersed at least one-tenth the diameter of the screw, and sufficient clearance should be allowed both below and at the side to prevent the

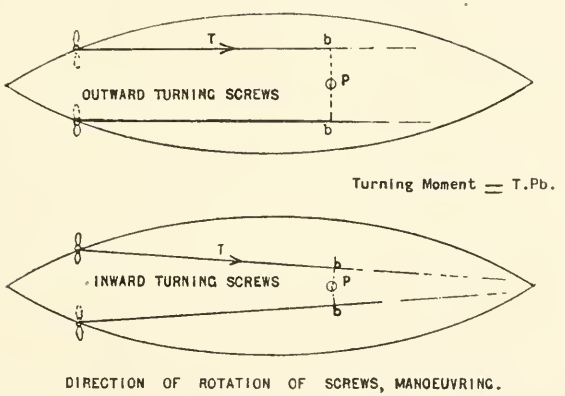


FIG. 371.—Manœuvring with Twin Screws.

screw being carried away by grounding or going alongside another vessel or pier. If a single screw is employed, the flow of water to it is necessarily masked by a portion of the stern of the ship to which it is attached. The area acted on by the screw is very nearly in proportion to its diameter; and taking this into consideration, it is evident that a greater amount can be acted on with two screws than with one. Even if only the same area of water column is acted on by two screws as by one of larger diameter, there is still the effect of greater immersion and better propulsive effect of the two screws. A space of about 1 to $2\frac{1}{2}$ feet is generally allowed between the tips of the blades of twin screws and the walls of the vessel abreast the screws.

The advantages of twin screws may be considered as :—

1. Propellers can be of smaller diameter, the total disc area can be increased, and there is greater immersion with less racing; for warships there is less chance of disablement by shot and shell and floating wreckage.

2. Better run of water to the screws, with a possible increase of efficiency.

3. Manœuvring power.

4. Steering by screws alone: one screw at a constant speed, and the other varied as necessary to keep a course.

5. Duplication in case of breakdown of one engine.

6. Division of engine rooms by a water-tight bulkhead, adding to the strength and safety of the ship. This is a doubtful advantage unless the bulkhead is athwartships.

7. The engines can be smaller and are thus under better protection.

Triple Screws.—When the beam is greater than three times the draught of water, the question of three or more screws must be considered if the power to be used and, therefore, the thrust to be imparted by the screws, cannot be obtained with twin screws. There appears to be no limit at present to the power which can be placed on one shaft by any available engine power within the limits of the vessel. The adoption of triple screws is therefore more dependent on the efficiency of the screw than on the possible efficiency of the engine.

Quadruple Screws.—This arrangement has the comparative advantage of twin screws over a single screw, and has been generally adopted in preference to three screws in the largest turbine-propelled vessels (see Turbines).

Cavitation.—The propeller acts by giving the water a sternward velocity, the thrust being equal to the change of momentum, which is equal to the acceleration multiplied by the mass of water acted on per second. It is found that about half the velocity is given to the water before it reaches the screw, and about half after leaving it. The particles react on each other, and the force required to accelerate each particle is supplied by the difference of pressure before and abaft it.

Consider consecutive particles, P . . . Q . . . R, just forward of the propeller: then the pressure forward of Q is the same as the pressure abaft P, and therefore less than the pressure forward of P by the amount necessary to accelerate P; and the pressure abaft Q is less than the pressure forward of Q by the amount necessary to accelerate Q. Thus the pressure decreases as the propeller is approached, owing to the consecutive acceleration of each particle.

The pressure just forward of the propeller, in still water, is that due to the atmosphere *plus* the head of water above the propeller; and when the propeller is moving, the total head is diminished by that necessary to accelerate the particles of water forward of the propeller—that is, the reduction of pressure just forward of the propeller cannot be greater than the sum of that due to the atmosphere and to the head of water above the propeller. Therefore, if half the thrust pressure per square inch on the blades (*i.e.* the equivalent of a reduction of pressure on the forward side of the blades) is equal to that due to the total head, the pressure just forward of the blades becomes *zero*. At *zero* pressure the water will "break," and cavities are formed forward of the blades. This is called *cavitation*.

Cavitation, principally due to centrifugal action, can also occur abaft the screw with a very high speed of rotation of the propeller.

To avoid cavitation, the following conditions must be fulfilled :—

1. The projected surface of the screw blades must be sufficient.
2. The screw must be sufficiently submerged.
3. There must be fine lines to give a clean run of water to the propeller; because a greater suction is required to draw the water to the propeller if the run be not sufficiently fine.

Thrust Pressure.—For any form of screw, and shape or form of vessel, there is some definite limit to the speed at which water reaches the propeller, and this limit may be considered to define the thrust pressure which can be imparted by the screw blades without causing cavitation. Cavitation, then, will take place, as we have seen, when the *projected area* of the blades is so small that the suction thrust—which may be taken as one-half the total thrust—per unit of area is greater than the atmospheric pressure *plus* the head of water above the screw.

The head of water may be usually neglected; thus the limiting thrust per square inch on the projected area may theoretically be considered as twice the atmospheric pressure, or 30 lb. per square inch. If the screw be sufficiently submerged to make the head appreciable, an addition of $\frac{3}{8}$-lb. per square inch can be made to the thrust pressure for each foot-depth of immersion of the tips of the blades. In practice, a thrust pressure of 30 lb. per square inch cannot be realised; for, of course, cavitation takes place before zero pressure is reached, on account of the air and vapour which is held in suspension or solution in the water becoming freed at a low pressure.

The practical limit of thrust pressure with screws of ordinary form

and shape, such as have been found to be efficient for fast-running engines and high speeds, is about 10 lb. per square inch on blades submerged about one foot. This thrust is that on the *projected* area of the blades ; the *developed area* depends on this projected area and the pitch-ratio adopted.

Surface Friction of the Blades.—The frictional resistance of the blades is of great importance ; it depends on the condition of the surfaces, their area, and the velocities with which the various units of their area cut through the water. It is evident that a square foot of blade area near the tip will cut the water at a much greater velocity than a square foot near the boss, and that the frictional resistance in consequence will also be greater. The form of blade usually adopted until recently was elliptical, with the minor axis about four-ninths of the major axis, which extended from the tip of the blade to the centre of the shaft. If the same blade area is used and the minor axis is increased to about six-ninths the major axis, the diameter of the propeller can be decreased. This alteration may produce a decreased frictional resistance, and at the same time an increased efficiency from an increased immersion of the tips. Some results tend to prove the practical efficiency of this alteration ; but it must not be forgotten that any diminution of diameter also diminishes the area of the column of water projected astern by the propeller, so that now the screw is generally made of the largest possible diameter which can be efficiently immersed and suited to the engine speed, while the area is disposed as near as possible to the boss, consistent with a fair curvature of the edges of the blade, as shown in Fig. 369.

In warships particularly, it is important that the blade area should not be unnecessarily increased, because the extra surface friction reduces the economy at low powers. In all cases the blades should be cleaned at every opportunity, and polished on the surface, to reduce the frictional resistance ; and an increase of over a knot in speed has been frequently obtained by proper attention to this detail when in dock. With the great speed of rotation now used for ships with turbine engines, the condition of the surfaces of the blades is of great importance.

Projected Area of Propeller Blades.—For the speed of vessel required, the I.H.P. or S.H.P. and the number N of revolutions per minute and the pitch of the propellers can be calculated from previous practice. The effective H.P. may be taken as about 50 to 65 per cent of the S.H.P. (call this percentage K), and the thrust pressure on each

square inch of projected area of the propeller blades is limited by practical considerations of cavitation, etc., to 10 lb. ; then—

$$\text{Total effective thrust} = \frac{\text{S.H.P.} \times 33{,}000 \times \text{K}}{\text{pitch} \times \text{N}} \text{ lb.}$$

$$\text{Projected area in sq. ft.} = \frac{\text{S.H.P.} \times 33{,}000 \times \text{K}}{\text{pitch} \times \text{N} \times 10 \times 144}$$

$$= \text{K} \times 23 \times \frac{\text{S.H.P.}}{\text{pitch} \times \text{N}} \text{ nearly.}$$

$$\left.\begin{matrix}\text{For turbine engines} \\ \text{(and S.H.P.)}\end{matrix}\right\} \text{K} = \text{about 65 per cent} = 0{\cdot}65.$$

Turbine-screw Propeller.—A type of Thornycroft turbine pro-

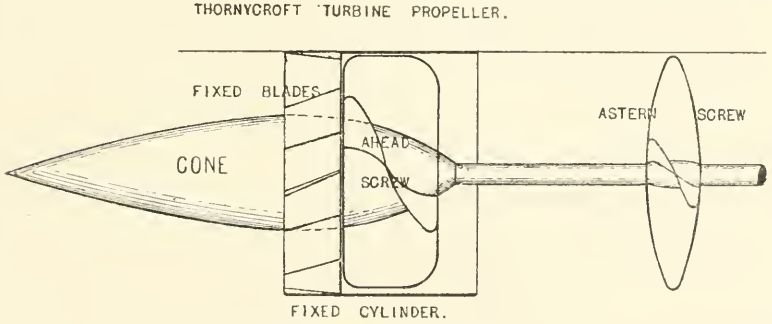


FIG. 372.—Thornycroft Turbine Screw.

peller is shown in Fig. 372, and is sometimes used for shallow-draught vessels.

The ahead screw is of small diameter, and revolves inside a cylinder fixed to the vessel. The revolution of the ahead screw, which is of increasing pitch, produces some rotation in the column of water passing through it. This rotation is reversed by fixed guide blades fitted inside the cylinder just abaft the screw, and the column of water is projected directly sternward without rotation. There is some loss of efficiency from the friction on the guide blades, but it appears to be small compared with the extra thrust obtained per unit of area of the blades. At the after end of the guide blades there is an elongated cone, of the same type as fitted to the ordinary screw propeller ; but it is generally longer, and the water thus takes up its sternward motion with the least possible disturbance and no eddying wake.

The pitch at the fore end of the ahead propelling blades corresponds approximately to the velocity of the water entering the screw, and the velocity of the water passing through is accelerated by the gradually increasing pitch of the blades towards the after end. As the flow of water is accelerated during its passage the area through which it passes is gradually decreased by the front part of the cone, through both the propelling and the guide blades.

This type of propeller cannot be efficiently used for going astern, because the column of water entering the guide blades from the after end cannot be sufficiently accelerated, and therefore little useful thrust can be imparted. For going astern another screw is fitted on the same shaft, as shown, at some little distance in front of the ahead screw, and of the same diameter and pitch as the forward end and

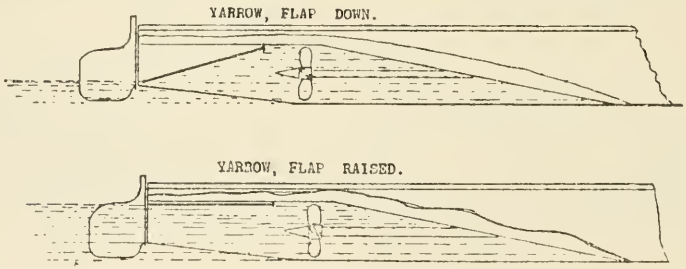


Fig. 373.—Yarrow Shallow-draught Tunnel.

leading edge of the "ahead" screw. When going "ahead" the astern screw imparts no thrust, and simply revolves with the shaft.

Mr. Barnaby estimated the thrust produced by the guide blades at about one-third of the total thrust of the propeller, but that the extra friction absorbed sufficient power to reduce the efficiency to a little less than that of the best ordinary screw. A thrust pressure of about 26 lb. can be allowed on each square inch of projected area of the turbine screw, with about the same efficiency as the ordinary screw propeller.

The turbine screw is not necessarily totally immersed when the ship is at rest. It can be fitted in a tunnel of suitable shape, so that after a few revolutions of the screw all the air is expelled, and the screw then acts in solid water.

Yarrow Shallow-draught Arrangement.—When a boat designed for shallow draught is loaded down considerably, the water passing through the propeller cannot easily escape, and there is con-

sequently a drag on the vessel, set up by the pressure exerted by the projected water on the after end of the tunnel. To overcome this difficulty, Sir Alfred Yarrow introduced an arrangement by which the after end of the tunnel can be raised or lowered. A horizontally hinged flap, partly forming the roof of the tunnel, is fitted just abaft the screw, and its trailing edge is raised or lowered as necessary to keep it at, or just below, the surface of the outgoing water. In Fig. 373 the flap is shown down in one view, as used in normal state of draught of water; and in the other the flap is raised for abnormal draught. A distinct and large gain in speed was obtained with this arrangement, when the boat on which the trials were made was loaded down considerably. The flap must be sufficiently raised when going astern, to allow a free flow to the propeller.

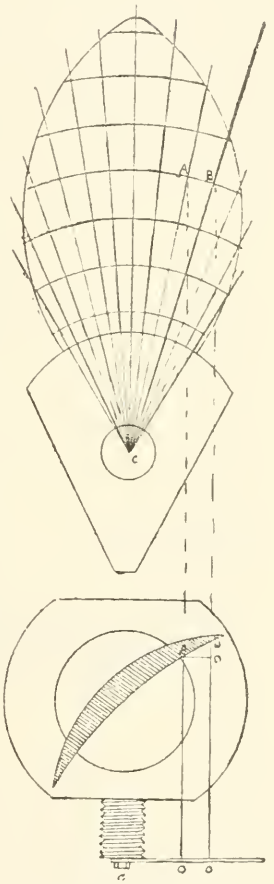


FIG. 374.
Measurement of Pitch of Screw.

Measurement of the Pitch of the Screw.—The pitch of the screw propeller is measured by placing the completed screw, set at about the required pitch temporarily, on a large face-plate, with one of the faces of the boss resting on the plate, or parallel to it. The driving face is preferably placed towards the plate, and then all measurements can be taken directly from it. In this manner a more perfect reading of the various pitches can be obtained than in the method generally used after the screw is fitted in place on the shaft, and after its measurement in this accurate way any small deviations can be removed by grinding the face to a truer surface.

When the screw is on the shaft in dock the method of measurement is as follows (Fig. 374):—The tail-piece at the end of the shaft is removed, and a radius bar, *CO*, fixed at the end *C*, fitted to the end of the shaft. The point *C* is made coincident with the centre or axis of the shaft, and when the bar is moved it

should always lie in a plane perpendicular to the shaft. A number of parts of circles, at different radii, pitched not more than one foot apart for large propellers, are described on the propeller blade by means of a bar, *OA*, with a sliding connection, which connection can be clamped at any required radius on *CO*. The bar *OA* must always be parallel to the axis of the shaft, so that the circle described from *C* as centre has *CO*, equal to *CA* in the upper figure, as radius. Denote any radius, *CA*, by *r*. By means of a circle on the end of the shaft or propeller boss, set off radii at equal angles. Call each of these equal angles, through which the radius bar *CO* is moved, θ.

The pitch is found for every point where the radii cut the circles ; and the *mean* of the pitches thus obtained is called the *mean pitch*.

Example.—Suppose the pitch is required at any point, A.

Then measure OA and OB accurately, from which referring to the lower figure, BD = OB - OA. In one complete revolution the point O describes a circle, the circumference of which circle is $2\pi r$, of which AD is a part. This circle is cut into $\frac{2\pi r}{\text{AD}}$ equal parts by the radii. Also in one revolution the point A travels, in a direction measured parallel to the axis, a distance equal to the pitch of the screw, which may be divided into $\frac{2\pi r}{\text{AD}}$ equal parts, each part being equal to BD.

Therefore the pitch at the point A (or practically at any part AB, if A and B are near together)—

$$= \text{BD} \times \frac{2\pi r}{\text{AD}}, \text{ where } \pi = 3{\cdot}1416.$$

$$\text{Or, the pitch} = \frac{\text{BD} \times 2\pi}{\theta}, \text{ where } \pi = 180^\circ, \text{ and the angle } \theta \text{ is known.}$$

For example, let $\theta = 7\frac{1}{2}^\circ$; then the pitch—

$$= \text{BD} \times \frac{360}{7\frac{1}{2}}$$

$$= \text{BD} \times 48.$$

That is, the pitch is 12 inches for every $\frac{1}{4}$-inch of BD.

A *more appropriate mean* is obtained by calculating the mean pitch at each radius and taking the mean of the means thus found. The greater the number of radii employed, the nearer will the approximation be to the correct mean. Generally, this second method of calculation after measurement will give a smaller mean pitch than that obtained from the previous method.

Chapman-Hunter Pitchometer.—This small instrument, which can be carried in the pocket, can be used for measuring the pitch of any sized propeller without calculation, either in the shop or in position

on the shaft. Lines are drawn at various radii on the blade from the centre of the boss, say at intervals of 1 foot. The instrument is shown in Fig. 375, and consists essentially of a plate, P (with feet, KK), on which scales are engraved for various diameters; an adjustable spirit level, L; an adjustable arm, M; and a thumb-screw, T, for adjusting the position of the level and at the same time securing it to the arm.

If the propeller be in the position shown in Fig. 376, the instrument is first placed in the position A with its feet resting against the face of the boss; the arm is kept closely pressed into its zero position, and at the same time the level is set until the bubble shows exactly at the middle line; and the level and arm are then secured by the thumb-

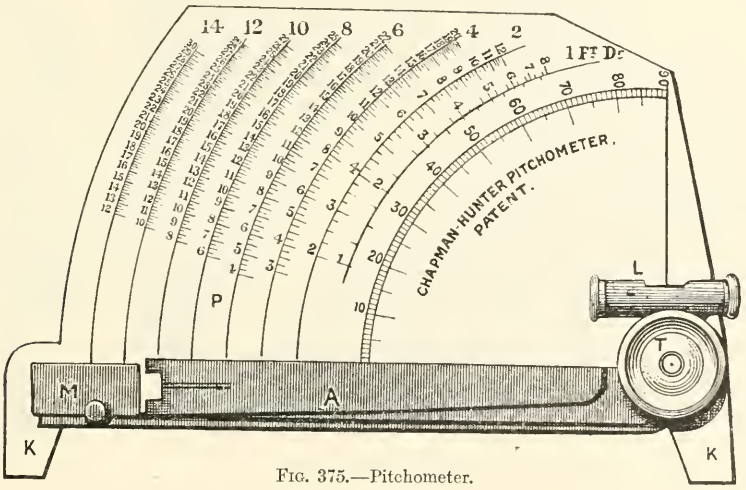


FIG. 375.—Pitchometer.

screw, so that they cannot alter their relative positions. Next, the instrument is placed at B with its feet resting on a pitch line; the arm is adjusted until the bubble again shows in the mid-position, and the pitch, corresponding to the cross lines at the edge of the arm and diameter of the pitch circle, can be read on the scale. For blades whose tips slope sternwards, and overhang the boss and tail-piece, it is convenient to turn the shaft as necessary to measure the down-coming blade, starting with the smallest diameter of pitch circle, and turning the shaft as necessary for each pitch circle until all are measured. In this way the pitchometer remains plumb and gives correct results.

Disposal of the Pitch.—There are three methods in general use of disposing the pitch :—

1. The most general is to use an absolutely true thread, with the pitch uniform throughout. This is commonly used in the Navy and the merchant service, but there are a few exceptions.

2. The pitch is made somewhat larger towards the following edge, and also increasing towards the tips of the blades. It is thought that a greater acceleration can be given to the water as it passes through the screw by this increasing pitch, and it appears a reasonable theory.

3. The pitch is decreased towards the tips of the blades with a view

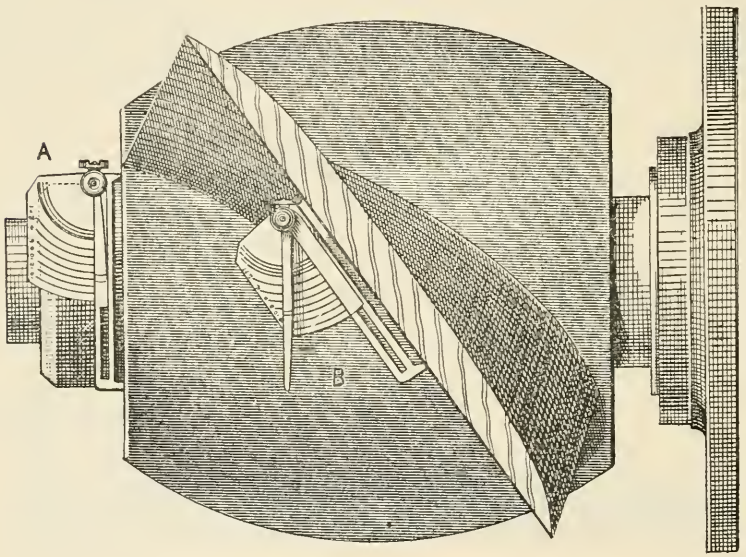


Fig. 376.—Application of Pitchometer.

to reducing cavitation, which generally commences at the tips of the blades.

At present there is insufficient evidence on any one of these to confirm its superiority to any other. In some large German liners the driving faces of the blades are machined to an absolutely true designed pitch, which is generally of the slightly increasing type. There is some advantage in this practice, but whether the advantage is due to increased accuracy or to reduced friction is a matter of some doubt. On the other hand, Messrs. Harland & Wolff, and probably other firms, trust to obtaining blades of exactly similar form by first making a cast-iron blade from the wooden pattern, and then using the cast-iron one as a pattern for the actual blades made of manganese bronze.

The surfaces of the blades are ground fair, and the leading and following edges are carefully chipped or filed to clean cutting edges. The surfaces are seldom painted, and the propellers give apparently good results. With fast-running screws driven by turbines or high-speed engines, a polished surface increases the efficiency, and incidentally it may be mentioned that this practice was initiated in the Navy by the author about 1895 in H.M.S. *Decoy*; and non-painting and a good polished surface is now considered the best practice.

PART IX

AUXILIARY MACHINERY

CHAPTER XXIX

THE STEERING AND CAPSTAN ENGINES AND GEAR

Introductory.—Most of the auxiliaries necessary to the working of the *Main*, or Propelling machinery have already been described with their uses, but there are others which, although they contribute nothing directly to this purpose are necessary for various reasons. The former are distinguished in the list below by the prefix M, and the others by the prefix A. The list is applicable to a large light cruiser launched in 1919, and capable of developing about 80,000 S.H.P. when burning oil fuel only.

In each of three boiler rooms:

M. 4 Forced Draught Fans, with a capacity of 55,000 cubic feet per minute at 450 revolutions per minute and $4\frac{1}{2}$-inch air pressure.
M. 4 Main Feed Pumps, each delivering 140 gallons per minute at 14 double strokes per minute.
A. 2 Auxiliary Feed Pumps of same capacity as Main Pumps.
M. 6 Oil Fuel Heaters.
M. 2 Oil Fuel Pumps, each capable of delivering 15 tons per hour at 21 double strokes per minute.
A. 1 Oil Fuel Hand-pump.
A. 1 Fire and Bilge Pump, 50 tons per hour at 25 double strokes per minute.
A. 1 Turbine-driven Salvage Pump, 1000 tons per hour.

In one of the boiler rooms only:

A. 1 Air Compressor for boiler tube sweeping, to maintain 50 lb. per square inch pressure through one $\frac{3}{4}$-inch nozzle.

In each of two engine rooms :

M. 2 Main Circulating Pumps, capable of pumping 19,000 gallons per minute against 38 feet Head at 400 revolutions per minute.
M. 4 Main Air Pumps.
M. 1 Direct Contact Feed-water Heater.
M. 2 Grease Extractors.
M. 3 Forced Lubrication Pumps each capable of delivering 10,000 gallons per hour against a Head of 100 lb. per square inch.
M. 2 Oil Coolers.
M. 1 Water-service Pump, capacity 20,000 gallons per hour at 24 double strokes per minute.
A. 1 Oil Fuel Service Pump, same capacity as those in boiler room.
A. 2 Fire and Bilge Pumps, as in boiler-rooms.
A. 1 Fresh-water Pump.
A. 2 Evaporators, 50 tons per 24 hours.
A. 1 Distiller with 346 square feet of cooling surface.
A. 1 Combined Circulating Water-Pump, Fresh-water Pump, and Brine Pump for distilling plant.
A. 1 Liming Tank and Connections.
A. 2 Electric Generating Sets, each 105 kilowatts, and 225 volts.
A. 2 Torpedo Air Compressors, each 30 cubic feet, 4-stage type. (In one engine room only).
M. 2 Steering Engines (in one engine room only), with 2 sets of Telemotor Gear for each engine.
A. 2 Turning Gear Electric Motors.

In other parts of ship :

Capstan Engines, 1 forward for working the anchor principally, and 1 aft for general purposes.
Refrigerating Engine, for food and vegetables.
Magazine Cooling Plant.
Workshop Motors.
Ventilating Motors.
Hydraulic Pumping Power Plants for working heavy guns.

In ships fitted with reciprocating engines, there is usually a small two-cylinder reversing engine for working the links of each main engine ; in the mercantile marine a single-cylinder direct-acting

arrangement is frequently fitted in conjunction with Brown's reversing gear.

Secondary Motors and Efficiency of Transmission.—Steam and oil-driven (or internal combustion) engines are the only *prime movers* fitted in sea-going ships, and all *secondary* motors, driven by compressed air, electricity, or hydraulic pressure, are derived from one of these primary sources. Therefore, in considering the application of any one of these secondary motors for any particular purpose, the over-all efficiency must be taken into account, and the comparison governed by the actual cost in fuel at the primary source of power. In the conversion of energy from the prime mover to the secondary motor there is always some loss, which is seldom less than 10 per cent, and is frequently much greater. Each particular case must be taken on its own merits, in which economy of fuel, weight, and space are of most importance, and convenience which may also be very important in certain instances.

In the boiler and engine rooms, steam engines are naturally the best and most efficient, and generally capable of accurate adjustment to the needs of the moment. Outside these spaces where long lengths of steam and exhaust pipes with their consequent great heat losses and heating effects in spaces in which such heat is not required, are generally undesirable, but for capstans, hoists, and winches on deck, their adaptability to immediate needs in speed and power give them a very decided advantage apart from their inherent efficiency, and for these purposes they are most frequently fitted. Electric motors are without flexibility in the power and speed required, and generally are heavier and occupy more space than the steam engine.

On the other hand, for torpedoes and their discharge, the best means devised depends on compressed air, and is generally inseparable from it. Heavy guns are more easily and accurately worked by hydraulic power than by any other means ; also a dominating factor is the ease of repair of damage which immediately shows its defects, and can frequently be at once localised and repaired. With electrically operated gun-mountings it is frequently difficult to locate a fault, and there is a danger of fire in undesirable places, in connection with the magazines.

For ventilation the electric fan has many natural advantages outside the boiler and engine room spaces. It is clean and generally free from heating effects in the compartments to be ventilated, and it can be run at a high rate of revolution with consequent good efficiency.

Although its efficiency is far less than a steam engine directly applied to refrigerating and cooling, the electrically operated cooling plant for magazines takes first place because it is a small apparatus and gives out no heat to contiguous compartments, one of which may be the magazine it is attempting to cool. This does not usually apply to the refrigerator used for cold storage of food and vegetables, and being a much larger unit also, a steam engine with its high efficiency and continuous working is usually best adapted for the purpose.

There are numerous pumps, both for main and auxiliary purposes, which can be run at a high rotational speed, and they could, and have been driven successfully by electric power, but never so efficiently as by direct steam power. For these the small turbine is now frequently used as motive power, and if the exhaust is conducted to some heat-saving device such as a feed-heater or evaporator, where it can do more work by giving up its remaining available heat, it is a very efficient method and compares favourably with any other combination of motors. Generally these small turbines save space and weight also and have a high rotational speed which is favourable in every way.

The steam and fuel consumption for harbour and auxiliary purposes is very great, and it is very necessary to economy that the engines from which secondary power is derived should be of the most economical in such consumption ; to obtain this generally more weight and space are required than are allowed, although it might and nearly always would save any increase in this way by its economy in fuel weight consumed in a few days' work. These remarks apply particularly to electric generators, which are great fuel consumers and constantly at work for lighting and power purposes. Turbines have advantages over reciprocating engines in that they will work well and comparatively economically on auxiliary exhaust steam when it is available ; at other times a condenser plant is necessary with an air pump to obtain economy, and it should be supplied and fitted in all vessels like warships which are in harbour for about one-sixth of their time.

Fig. 377 shows diagrammatically to scale the enormous proportion of heat in the steam exhausted from auxiliaries, which is unfortunately frequently wasted, but can be saved and utilised to heat the feed-water in a well-designed system. The few auxiliaries shown in this diagram are those generally found in cargo vessels, with special reference to Figs. 323 and 335 ; as pointed out above, the number of auxiliaries is sometimes very great.

Steering Mechanism.—Some mechanism for controlling the helm is an absolute necessity for large or fast vessels. The pressure on the rudder is very great when the ship is turned, and, without mechanism, is beyond control by manual labour. The time required to move the rudder is proportional to the power which can be employed in

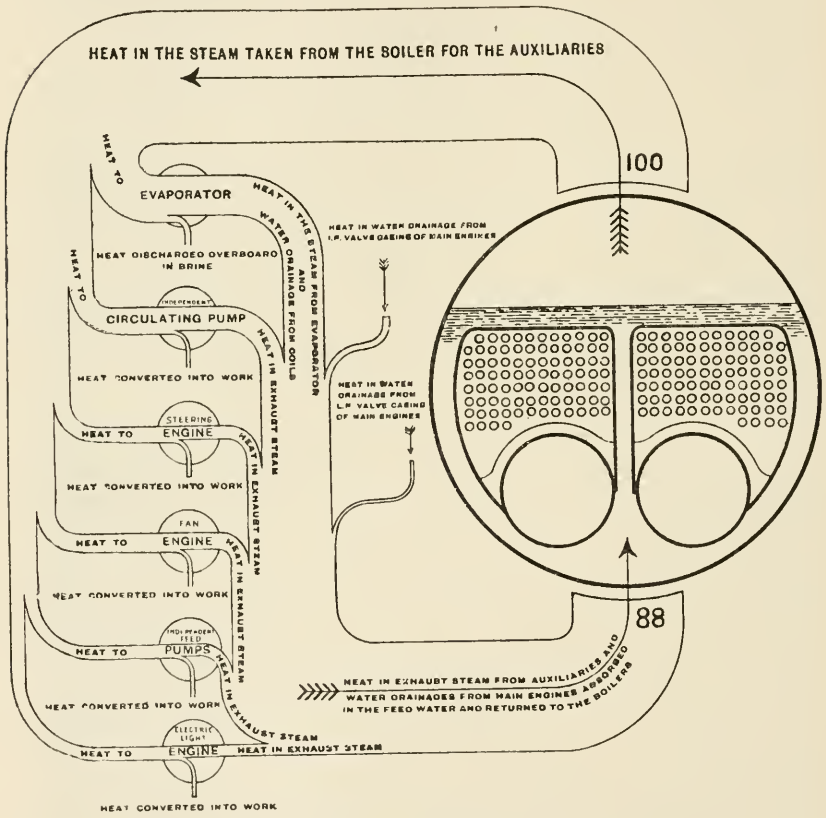


Fig. 377.—Utilisation of Exhaust Steam.

moving it, and consequently the first consideration, in designing a mechanism for the purpose, is to meet the time requirements from midship to extreme helm (about 10 seconds), and from extreme helm to extreme helm (about 20 seconds). The mechanism is generally constructed in three principal parts :

1. The controlling gear, worked by the helmsman at any required

point in the ship, which works a valve of the differential reversing type.

2. The engine, which obeys the movement given by the helmsman, through the controlling gear, to the differential reversing valve.

3. The mechanism or gearing, which conveys the power of the engine to the crosshead or tiller, rigidly connected with the rudder.

Controlling Gear. — A wheel is generally used for moving the controlling shafting, but in a few cases a small lever is used. The lever, or *upper* part of the wheel in naval steering gears, is arranged to move in the same direction as the bow of the ship when it responds to the movement of the rudder. The wheel mounted on a horizontal axis, *A*, within a vertical pedestal, *B* (shown in Fig. 378), rotates a vertical shaft, *C*, through bevel gearing wheels, *DD*. The shafting is continued by horizontal and other necessary lengths, connected by bevel wheels, or Hooke's joints, with a lever in connection with the differential valve spindle.

The number of turns of the wheel is limited, generally four revolutions from amidships to extreme helm. Inside the pedestal the shaft is screwed with a properly pitched thread, *E*. A certain number of turns of the steering wheel revolves the screwed shaft through a nut, *F*, which is moved along, but does not revolve with the shaft. A projection is made on the bevel wheel at *K* and on the collar *G*; and the wheel *DK* and collar *G* being keyed to the horizontal shaft, revolve with it. At the extreme points of the helm allowed, projections on the nut *F* come in contact with the projections on *DK* and *G*, and further rotation is prevented. A cogged rack, *R*, made in one piece with the nut *F*, moves with the nut and rotates a cogged wheel, to which a pointer, *P*, is attached; the pointer *P* is outside and generally on top of the pedestal. Two dials, one engraved on the top and the other printed on glass (so that a light placed inside the pedestal shows through it), show the amount of helm, in turns of the wheel and in degrees, imparted by the controlling gear. This index does not necessarily show the exact position of the rudder or tiller in connection with it; this is shown by the *helm indicator*, which is mechanically geared to the rudder head. If the controlling gear, engine, and rudder are properly connected to work in unison, then the pointer on the steering pedestal indicates the position of the *rudder*, as well as the controlling gear.

The details in connection with the controlling shafting are of great importance, as the safety of the ship frequently depends on its efficiency.

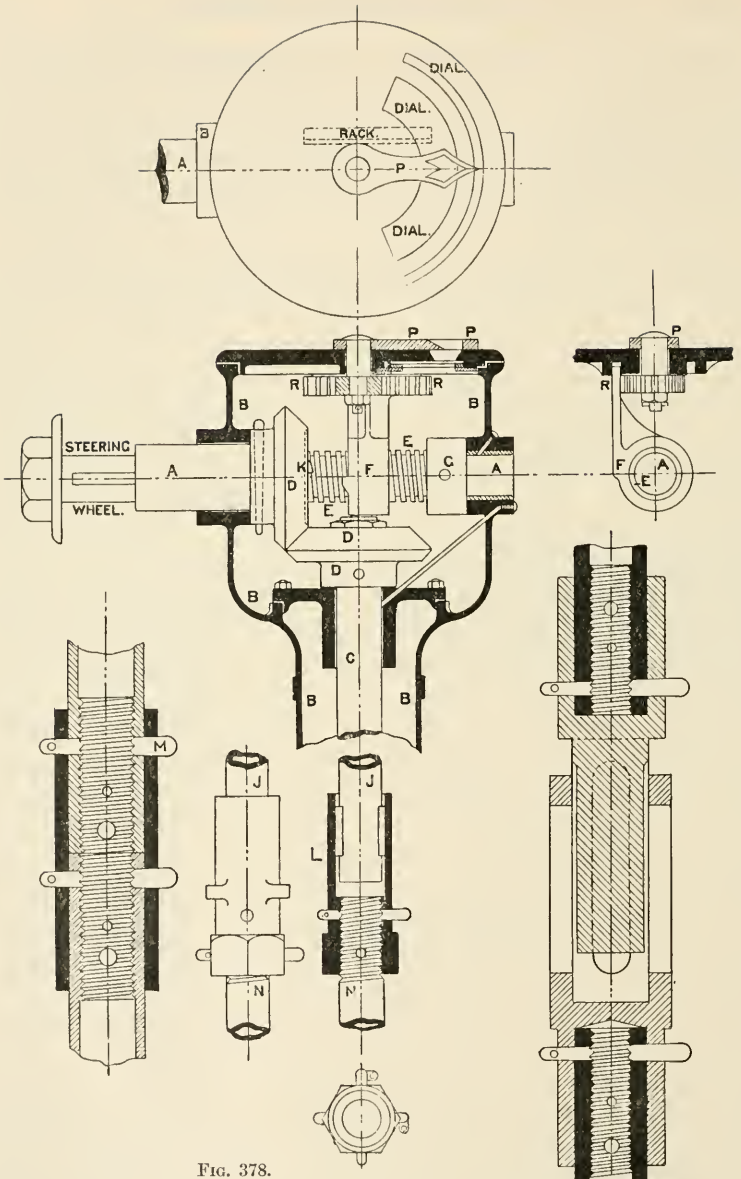


Fig. 378.

Fig. 379.

Figs. 378 and 379.—Steering Control Pedestal and Shaft Couplings.

Generally the shafting is hollow and stiffened at the couplings by screwed plugs, fixed by rivets, as shown in the various figures. The weight of the shafting, if horizontal and of any length, is carried on roller bearings ; the bevel wheels are carefully machined to gear into each other with as little back lash as possible, and without undue friction ; and the keys and pins securing the wheels to the shafting are fitted with stops or split pins to prevent them working out of place.

The various lengths of shafting are connected together by sleeve pieces, secured either by taper pins, *M*, or by cotters, or by small bolts and nuts, with split pins to prevent them falling out. The cotters at various parts of the shafting are free to move through elongated holes in the sleeve piece, and thus allow for expansion. In some cases squared ends, working in squared sockets, are used for expansion couplings ; but some stop should then be fitted to prevent the two parts of the shaft disconnecting under a sudden stress or alteration of form of the vessel. In a destroyer, such disconnection has taken place in very rough weather. Unless the deviating angle of the shafting is very small, bevel gearing is used in preference to a Hooke's joint. When steam is on the engine, a movement of less than two spokes of the steering control wheel, or one-eighth of a turn, should set the engine in motion.

If the steering wheel is run over very quickly, the engine following in unison with it, the momentum of the engine and the heavy parts moved by it are liable to carry the tiller over farther than the stops inside the pedestal would otherwise allow. The stops inside the pedestal are therefore generally set two or three degrees within the extreme range of the tiller. A connection is fitted to the rudder head or tiller, as convenient, which, coming against projections attached to the ship, acts as a stop in each of the extreme positions—generally about 35° each way from the hard-a-port or hard-a-starboard position to the middle position.

In warships, pedestals and steering wheels are fitted in various parts of the ship, and each is generally connected by shafting and a coupling with one of the main runs of shafting from the foremost steering positions. As far as possible the controlling shafting is kept below the water-line, under protection from shot and shell, and the couplings for the various steering positions are therefore generally in this position. The end, *J*, of the controlling shaft leading from any steering position, *C*, is generally connected with the main run of shafting by screwing into a sleeve permanently attached by a sliding joint, *L*,

to the main run by a short length of shafting, N, and bevel wheels. When the two parts of the shaft are in coincident positions with the same helm, as shown by their respective dials at the steering positions, the position of the end of the shaft is such with respect to the sleeve that it can be connected with it by inserting two taper pins, prevented from working out by split pins. This arrangement is necessary to make it impossible to connect up the various steering positions in

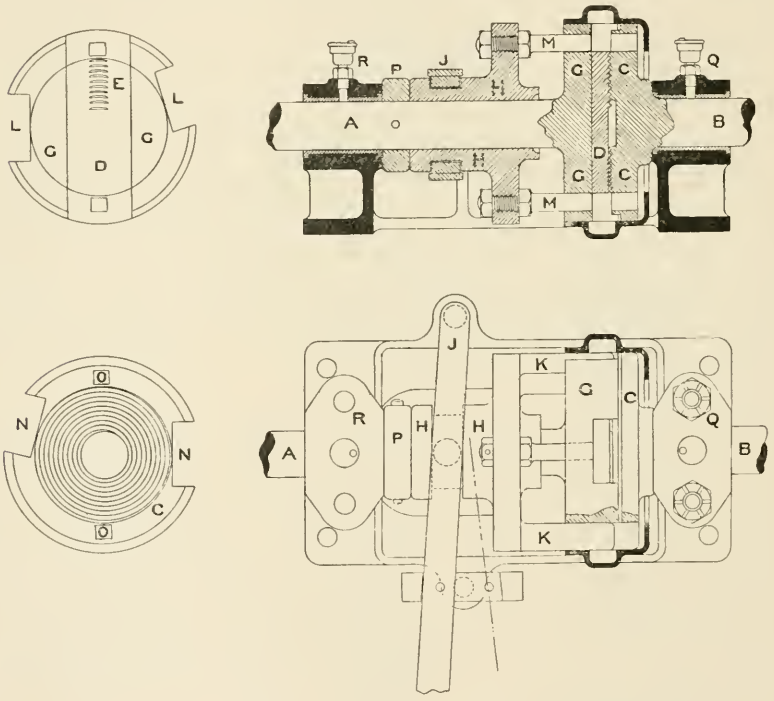


Fig. 380.—Brotherhood Coupling.

incorrect positions relatively to each other, and they can thus be connected correctly when communication with the various positions is impracticable.

Brotherhood Coupling.—The object of this coupling is to enable two shafts to be coupled or uncoupled at any time when one or both are being rotated ; the same end is attained in the screw coupling previously described, but the Brotherhood arrangement is handier and more convenient. Referring to Fig. 380, A and B are the two

shafts to be coupled. A flange, *G*, is made on *A*, and in a slot made in the face of *G* there is a sliding piece, *D*, with projecting teeth, *E*, which is constrained to move in the slot. The teeth *E* engage in a spiral projecting thread formed in the face of the opposite flange *C* of the shaft *B*. The figures show the uncoupled position; and when either *A* or *B* is rotated independently of the other, the sliding piece *D* moves to and fro in the slot corresponding to the motion imparted by the projecting teeth *E*. *H* is the coupling, keyed to the shaft *A*, and capable of being moved along *A* by the hand lever *J*. On *H* there are two projecting lugs, *K*, always engaged in the spaces *L* formed in the rim of the flange *G*, and which when the lever is pushed to the right also engage in the spaces *N* of the flange *C*, and thus form the coupling between *C* and *G*. The sliding piece *D* prevents the coupling being moved to the right, except when in the correct coupling position, by preventing the pins *M* fixed in the coupling from being moved to the right. When the shafts are in correct relative position the pins *M* enter holes, *O*, near each end of the sliding piece *D*, and almost simultaneously the lugs *K* engage in the opposite spaces *L*, thus completing the connection. The spaces *L* are not diametrically opposite each other, so that the connection cannot be made half a turn out of the correct position. The teeth *E* and spiral thread are always engaged, and longitudinal movement of the two shafts, which would throw them out of gear, is prevented by a shoulder on *C* and a collar, *P*, fixed to the shaft *A*, bearing against the ends of the shaft bearings *Q* and *R* respectively.

Brown's Hydraulic Telemotor.—In some cases small pipes conveying hydraulic pressure have been substituted for the mechanically geared controlling shafting. The arrangement consists, as shown in Fig. 381, of a hand-steering wheel which, through a series of cogged wheels, *G*, *H*, *I*, *J*, *K*, and shaft, *E*, and pinion, *D*, moves a cogged rack, *C*, attached to a piston, *B*, working in a cylinder, *N*, at the upper part of the pedestal *O*. The piston is packed with double leathers, and when in the middle of its stroke, free communication is allowed, through an enlargement, *M*, of the cylinder, between the upper and lower sides of the piston. The cylinder is kept fully charged with water, mixed with about 30 per cent of glycerine, to prevent it freezing. Pipes lead away from the top and bottom of the cylinder to another cylinder, *R*, called the *motor* cylinder, fitted near the steering engine. The piston *S* in the motor cylinder is connected with the differential reversing valve. Any movement of the wheel tends to compress the

water in one end of the first cylinder, and, forcing it through one of the connecting pipes, moves the piston in the second cylinder, and thus moves the differential valve and sets the engine in motion. The

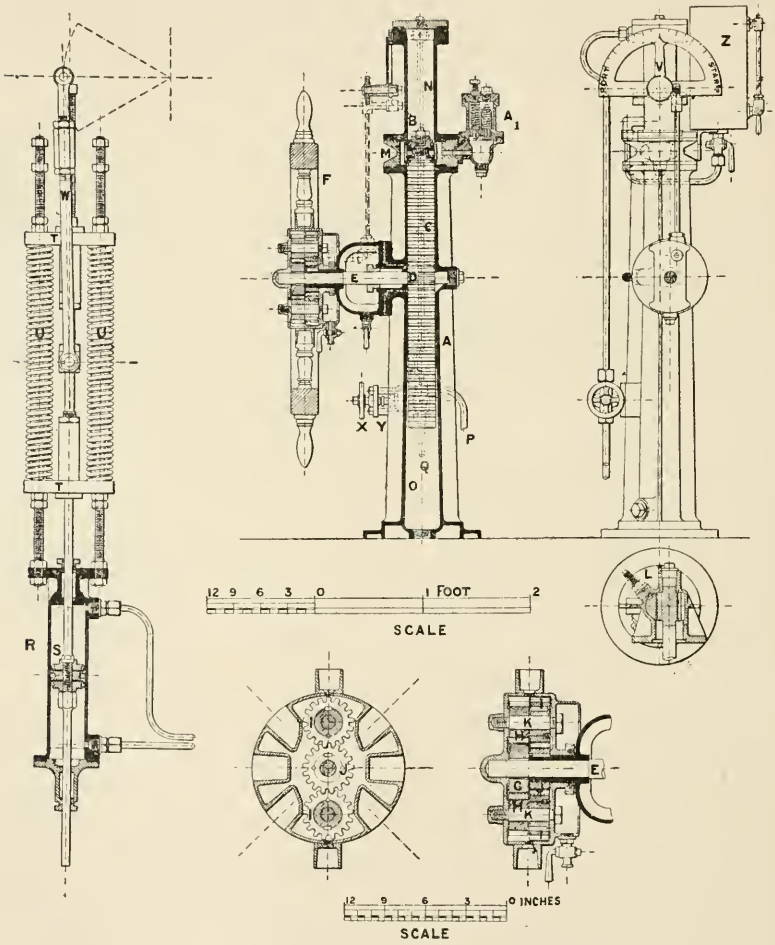


Fig. 381.—Hydraulic Telemotor Controlling Gear.

water expelled from the second cylinder is returned to the first cylinder through the other connecting pipe. The resulting motion of the engine, when the differential valve is moved, returns the latter to its central position and stops the engine, as in other types.

When the steering wheel is put amidships, free communication is allowed between the top and bottom of the piston in the first cylinder, as already explained, and the piston in the second cylinder is returned to the middle of its stroke by two springs, *U*, either one of which is compressed when the piston is moved away from the middle of its stroke. Consequently, whenever the steering wheel passes the amidships position, any inaccuracy between the relative positions of the two pistons tends to be adjusted. The first cylinder is twice the capacity of the second, and there is always sufficient head to overcome the frictional resistance of the passages.

Differential Reversing Valve.—By reference to Figs. 189 and 191, it will be noticed that when the crank and eccentric arms are in the positions *OP* and *OS* respectively, the direction of rotation of the shaft is the same as that of the hands of a clock. Now, consider the direction of motion when the steam is admitted through the exhaust cavity, and that exhaust takes place at the outer edges of the valve; then the direction of rotation is reversed, and becomes anti-clockwise. The valve used to obtain the changing over of the ports of entry and exhaust of the steam is called a differential reversing valve, and is the middle one of the three valves shown in Fig. 382, which are shown as cylindrical valves with an inner hollow space connecting the spaces at the ends of each respective valve.

The engine is fitted with a pair of cylinders (see also Fig. 383), and the two cranks are set at right angles on the shaft, so that there is continuous motion and the engine can be started from any position of rest. Steam enters the differential valve chest through a port, *S*, and the exhaust steam leaves by the ports *E*, which are connected with each other and the exhaust pipe. The movement of the steering control shaft operates on a pair of bevel wheels, one of which rotates the differential valve rod by means of a feather. This rod takes the place of the lever *ABC*; and its rotation either screws it farther into or out of one of a second pair of bevel wheels, which rotate in a fixed bearing and with the crank-shaft respectively. Suppose the movement of the control shaft has placed the differential valve in the position shown in the upper figure; then the engine is set in motion and revolves the crank in the direction shown by the arrows, and, rotating the upper pair of bevel wheels, the rod and valve are brought to the central position by the rotation of the crank, and the engine is brought to rest by the automatic cutting off of the steam-supply. Similarly, if the control shaft be rotated in the opposite direction, the crank-shaft is

also driven in an opposite direction and automatically cuts off the steam-supply and stops the engine.

The gearing which effects this automatic stopping is known as *Hunting Gear*, and is commonly fitted for steering engines in connection with the differential reversing valve ; this valve is used for many

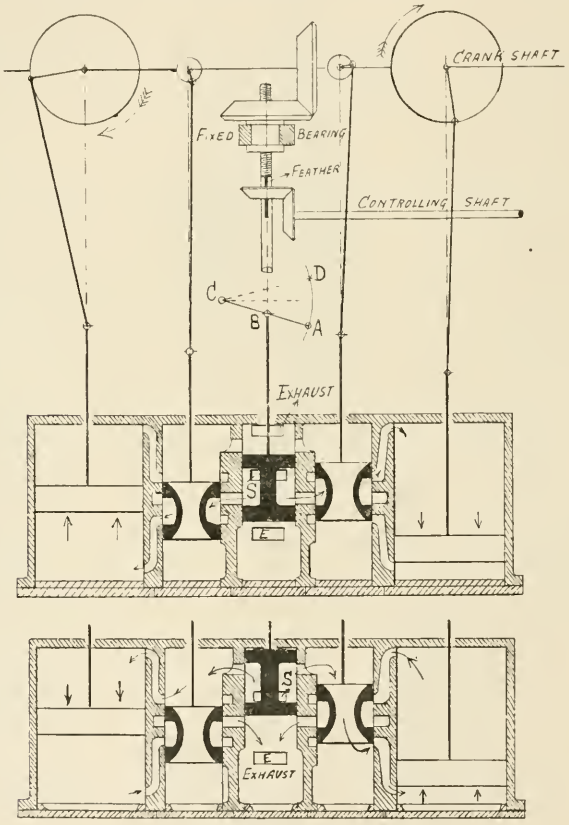


FIG. 382.—Hunting Gear.

engines in which economy is of little importance compared with a quick and certain movement and reversal. The principal ones are the steering and capstan engines, hoisting engines generally, whether driven by steam or water pressure, and the turning and reversing engines used in connection with the main propelling engines ; these engines are only occasionally, not continuously, in use for short periods, and economy is sacrificed to obtain practical utility.

Steering Engine.—In Fig. 384 three views in part section are shown of one of a pair of steering engines fitted in recent warships; each engine is an exact duplicate of the other, and by suitable couplings either one can be used as required. Each engine has two cylinders, *H*, with cranks fitted at right angles to each other on the shaft *A*. The differential reversing valve *F*, and slide valves *G*, are of the solid type, with open communication between the respective opposite ends of each valve through passages in the casting. Steam enters through

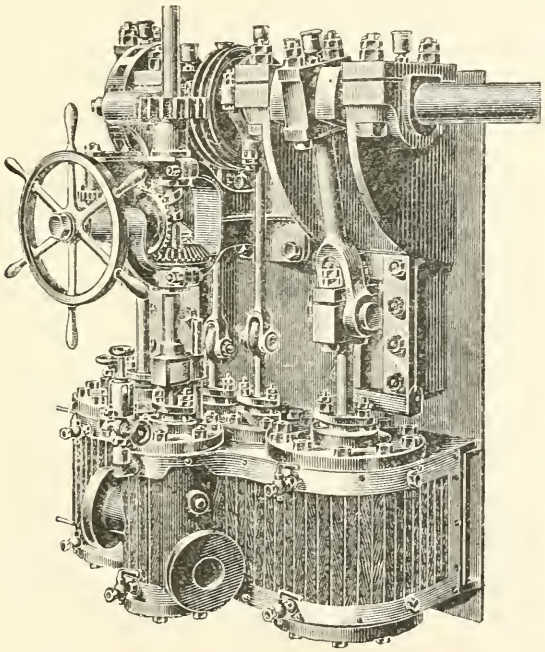


FIG. 383.—Steering Engine.

the pipe *S* and surrounds the middle part of the valve *F*. Exhaust steam leaves through the passage and pipe *E* and fills the spaces above and below the valve *F*, which is moved up or down by the rotation of either the control shaft *C* or the hand wheel *N*; *C* and *N* are usually connected, and rotate together. Rotation of the shaft *CN* screws it through the centre of the worm wheel *D*, which revolves in a fixed bearing, *Q*, and raises or lowers the valve rod *B* and thus admits steam to the slide-valve chests and sets the engine in motion. By means of a worm on the shaft *A* gearing in the wheel *D*, the rotation of the shaft

automatically shuts off steam, through the valve F, and stops the engine.

The framing P of the engine is usually secured to the engine room, or other, bulkhead. At M there is a hand-turning wheel for use when steam is not on the engine. K is an index (showing the relative position of the steering control wheel), which is moved by the spur gearing L and rotation of the control shaft.

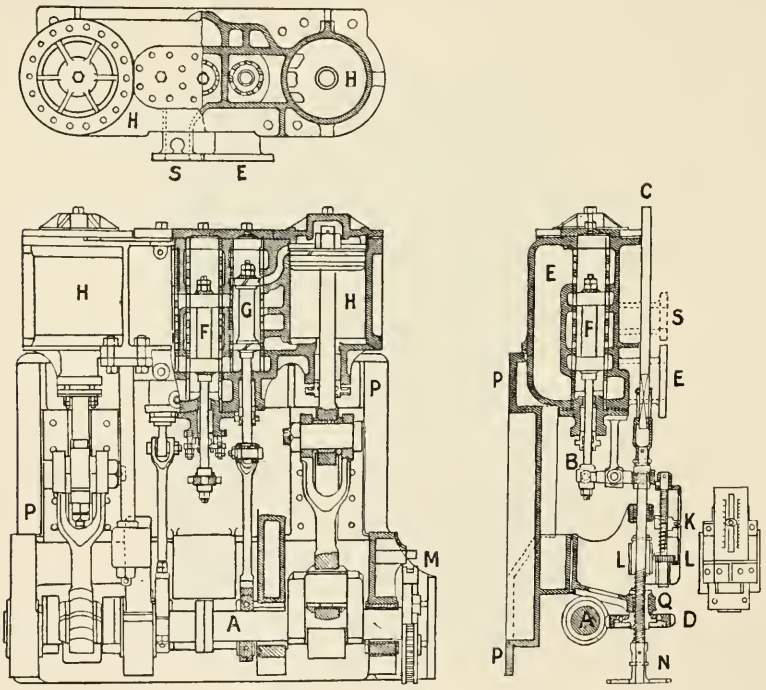


Fig. 384.—Steering Engine and Differential Valve Gear.

A slight lap is necessary for the differential and slide valves to prevent leakage, and consequently the steering control shaft and wheel must be moved a little before a cylinder is supplied with steam and the engine begins to move; generally one-eighth of a turn overcomes the back lash in the gearing and the effect of lap.

Rapson's Slide.—In this arrangement the tiller is worked by wire ropes or chains, or by pitch chains running over spiked wheels. The rope or chain is worked by the engine, and the two ends are connected at opposite sides of the end of the tiller with a carriage or slide.

The chain forms an isosceles triangle, with an angle below a horizontal base ; the chain is led over a pulley at each angle next the base, and, running horizontally, connects with the carriage and completes the base line of the triangle. The carriage either moves along a sunk slide or moves along on rollers attached to itself. The end of the tiller projects through a swivel block in the carriage, and as the tiller is moved over, the end slides to and fro through the block.

This gearing has the mechanical disadvantage of not increasing the force applied to the rudder in proportion to the greater degree of helm imparted by the engine ; an ideal arrangement should have an increasing effect. Wire ropes are still used for steering, and chains are common in small vessels, but are generally less reliable than the arrangements next described.

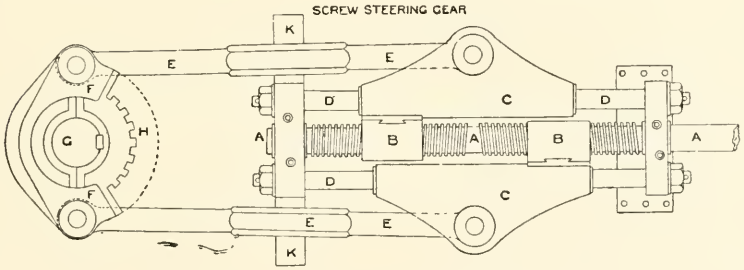


FIG. 385.—Napier's Differential Screw Gearing.

Napier's Differential Screw Gearing.—A plan of this gearing, which is fitted in a great number of vessels, and has been in common use for over fifty years, is shown in Fig. 385. The movement of the engine rotates the shaft *A*, on which square-threaded right- and left-handed screws are cut. The screws work in gun-metal nuts, *B*, attached to arms, *C*, sliding along fixed bars, *D*, and connected by rods, *E*, with a crosshead, *F*, attached to the rudder head *G*. The rotation of the shaft moves the nuts either closer together or farther apart, and thus moves the rudder. This gearing is self-holding, and no stops or pawls are necessary.

Sufficient end-play must be allowed for the screwed shaft in its bearings, and in its connection with the engine shaft, for the variation caused by obliquity. Thus when the arms are brought together, putting the rudder to port, the rod in connection with the left arm should be shorter than the other rod, and *vice versa*.

Harfield Steering Gear.—The engine is of the usual type, and drives a shaft connected by gearing with a vertical shaft, *A* (Fig. 386), on which a cogged wheel, *B*, is mounted *eccentrically*. The cogs on *B* engage with similarly pitched cogs on a rack, *C*, which is in rigid connection with a crosshead, *F*. When space permits, the crosshead *F* is fixed on to the rudder head directly; but generally space does not admit of sufficient movement of the rack, and the crosshead *F* is connected with the crosshead on top of the rudder by rods, *GG*; and *D* is simply a vertical shaft on which the rack *C* and the crosshead *F* are

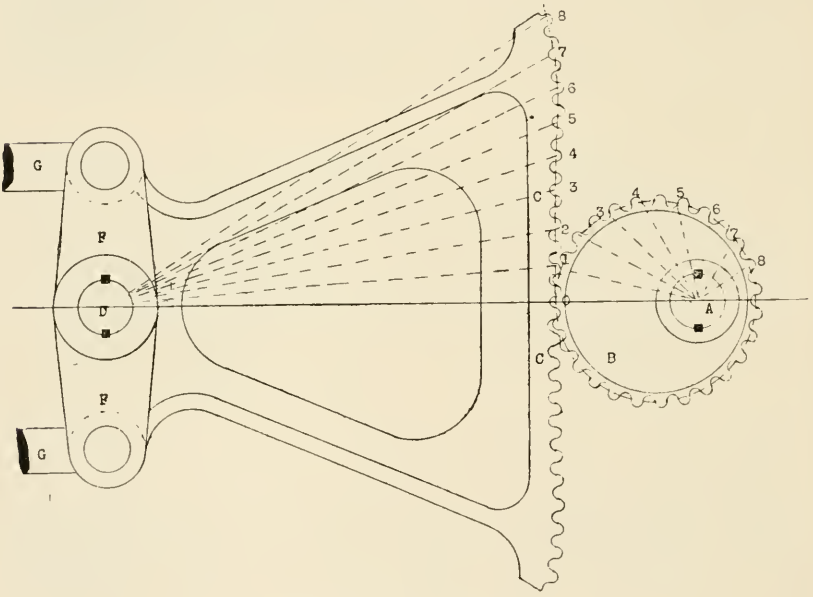


Fig. 386.—Harfield's Gearing.

mounted. The curvature of the rack is obtained graphically, so that when each of the numerals, shown on the wheel and rack, come into position on the fore-and-aft line joining the two centres of the shafts *A* and *D*, the cogs are properly engaged.

The advantage of Harfield gearing is that the force or turning moment applied to overcome the resistance of the rudder is increased when the rudder is moved farther away from its middle position. Thus at the middle position the ratio of the force to the resistance is as *DO* is to *AO*, and in the extreme position the ratio is as *D*8 is to *A*8.

If a worm screw is used in the connection of the engine shaft with the shaft *A*, the gearing is self-holding; that is, if the shaft breaks at some point previous to the worm shaft, the gearing holds the rudder in one position. But if an ordinary series of bevel wheels is used, without any worm screw, the gearing is not self-holding, and leaves the rudder free to move to and fro. Some arrangement is then sometimes fitted to hold the rudder in case of accident, and a rod working a piston in an oscillating cylinder is attached to one of the pins in the

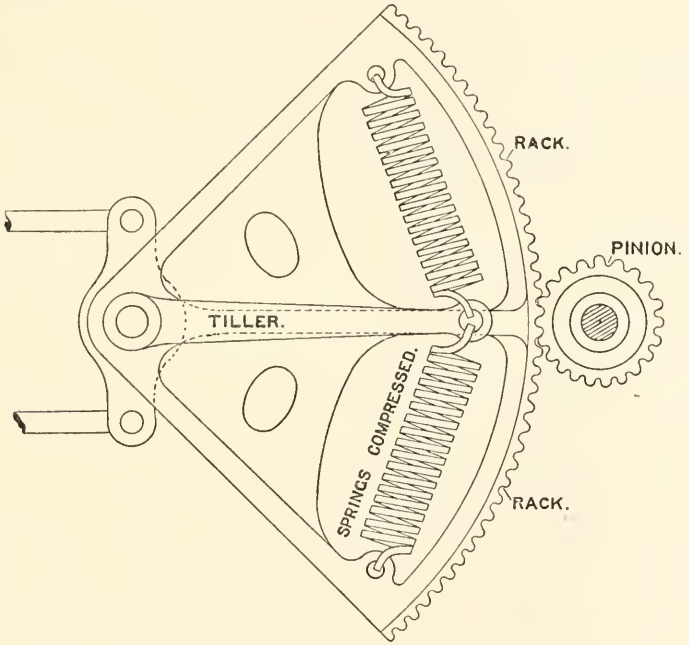


FIG. 387.—Wilson and Pirie's Rudder Gear.

crosshead *F*. The cylinder is used as a buffer, and can be filled with water under pressure.

Wilson and Pirie's Gear.—In this gearing the pinion is fitted *concentric* with the shaft, the rack is circular (generally a quadrant), and motion is imparted from the rack to the tiller through strong spiral springs.

The tiller is separate from the quadrant, but pivots about the same centre, and is connected, through a hole in the quadrant, with the rudder, or false rudder, head. The springs are always *under*

compression, and take any shock caused by alteration of helm or motion in a sea-way. The gearing is thus compensated for back lash, and generally noiseless.

Steering Transmission Gearing.—The engine power of the steering engine is necessarily very great, not alone by reason of the great pressure on the rudder which it has to overcome when turning the ship, but also by reason of the very low transmission efficiency of the gearing and shafting between the engine shaft and the rudder head. In some instances, particularly where the rudder is dependent on internal (to the ship) pintles and is unsupported by a pintle at the base, the frictional resistance is very largely increased by the enormous pressure exerted on the internal pintles.

From an engineering point of view, and for simplicity and efficiency, many of the arrangements of steering engine and its contingent gearing are very poor examples, and in a few instances which have come under the author's notice, may be considered as complicated freaks of low efficiency, and some appeared to be dangerous in a sea-way. Simplicity and efficiency should be the objects aimed at, and not the least in importance is a safe arrangement in a sea-way for adjustment and repair, or, in case of emergency, the substitution of hand-operated steering. With practically unlimited space in mercantile vessels, these desirable features should be easy of attainment.

In warships, protection from gun-fire is one of the principal considerations, and in some vessels of the destroyer type external pintles, at or near the base of the rudder, are undesirable. The steering engine is frequently fitted in the main engine room, and in many instances is duplicated in the larger vessels. In destroyers there is usually only one engine fitted on the after engine room bulkhead which transmits its power through worm and screw gearing, and a long fore-and-aft shaft to the tiller compartment above the rudder. Several shaft bearings are necessary, and in many cases one or two knuckle-joints of the Hook joint type are fitted to allow for any alteration of form of the ship in a sea-way. In the tiller compartment the shaft is connected through spur or worm gearing with the differential gearing, which is usually of the self-holding type, known as Napier screw gearing.

Eventually the *force* available for turning the rudder head is probably less than 10 per cent of the force exerted by the engine shaft. In a typical arrangement the author made a very careful estimate, and found the over-all efficiency about 8 per cent. This was early in

1915, when reports of great trouble with jambing of the rudder in many destroyers were prevalent, and in consequence of which he devised a better system of lubrication for the rudder head itself, as this appeared to be the seat of the trouble. By substituting a horizontal circular groove (Fig. 387A) in connection with the oil pipe entering the vertical bearing immediately below the packing gland, for the usual zigzag (vertical) groove generally cut from the pipe hole down to the bottom of the bearing, which incidentally allowed the oil to run into the sea or alternatively allowed the sea-water to flood the oil pipe and lubricator box, a great improvement was effected in the efficiency of lubrication ; and then, by substituting a closed lubricator with a screw punch action in connection with the original pipe, a supply of thick oil or grease could be forced into the horizontal groove, from which it gravitated down the whole internal face of the bearing. This alteration improved the life of the bearing from months to years, because when the clearance exceeded about $\frac{15}{1000}$ths of an inch, renewal was necessary or jambing occurred. This jambing was particularly prevalent in the early actions in the North Sea, but by May 1916 the alteration made to the lubricating arrangement had so improved matters that not a single destroyer of the many engaged in the Battle of Jutland experienced any trouble from jambing, or from the steering gear generally.

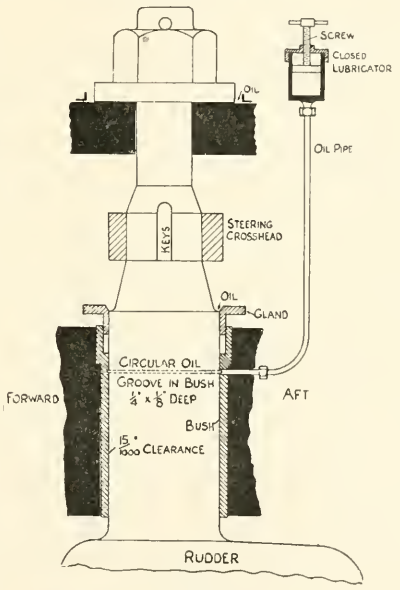


FIG. 387A.
Tompkins' Rudder Head Lubrication.

The application of the principle of supplying lubrication through a horizontal groove to all vertical spindles of capstans, derricks, etc., would be a great improvement on the usual system of staggered (vertically arranged) grooves, which waste oil without providing efficient lubrication.

Capstan Engine and Gear.—The capstan engine and gear are

used for getting up the anchor, and catting it if necessary, and for warping or hauling the ship from one position to another. The engine is usually of the two-cylinder type, with two cranks at right

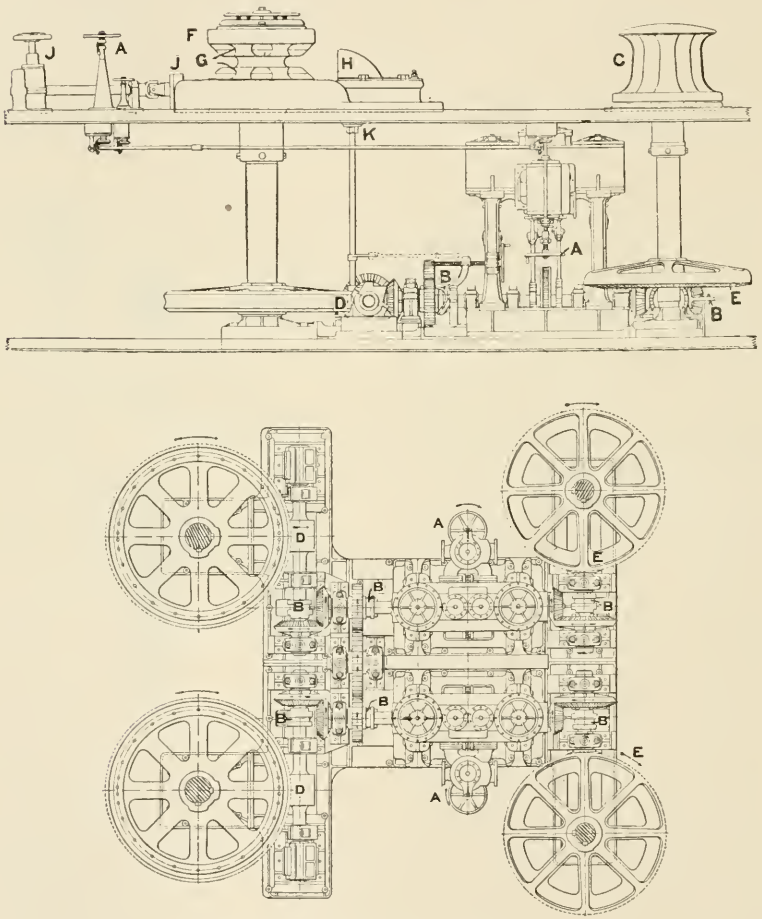


Fig. 388.—Napier's Capstan Engine and Gear.

angles on the shaft, and it is controlled by a differential valve similarly to the steering engine.

In Fig. 388 an elevation and plan of the general arrangement of two engines, two cable holders or windlasses, and two capstans are shown, as fitted by Messrs. Napier Brothers in several large warships

and mercantile vessels. In warships the engines and gearing are fitted below the armour and deck protection; this entails longer spindles and control gearing shafting, but otherwise the arrangement is substantially the same as shown in the figure. *A* is the control wheel and shafting, which can be worked from the forecastle deck or at the engine. By means of clutches and couplings, *B*, either engine can be used in connection with either cable holder, *F*, or capstan, *C*. The two capstans are used ordinarily for hauling or warping, and, where necessary, for catting. The two cable holders are used almost exclusively for weighing the anchor, and due to the worm screw *D*, instead of bevel pinion, *E*, gearing move more slowly but exert greater force. The cable is passed half round the holder *F*, fitted with stops, *G*, to obtain a good grip, and then falls through the navel pipe *H* into the chain locker below. The cable holder can be held in a fixed position by a friction brake, *J*, of which there are several varieties, and the cable is held by the compressor *K*. The compressor is now usually made as a lever pivoted horizontally to one side, close up and below the navel pipe. When not in use, the cable is free to run through a hole (of the same diameter as the navel pipe) in the lever; by pulling the lever to one side, using a tackle for the purpose in large ships, the cable is "bowsed-to" by closely nipping it beween the compressor and the edge of the navel pipe. As additional securities, the cable is usually "bitted" and a "slip" fitted.

Relative Stress in Cable and Steam Pressure in Cylinders.—Of the indicated work done on the engine pistons, a reduction is caused respectively by the engine friction, friction in the bevel gearing (see Fig. 388), the worm screw gearing, the cable holder spindle, and of the cable moving over the deck, etc., and through the hawse pipe. For ordinary purposes of calculation, the respective efficiencies may be taken as, $\frac{4}{5}$, $\frac{4}{5}$, $\frac{1}{2}$, and $\frac{4}{5}$; and the over-all efficiency—

$$E = \frac{4}{5} \times \frac{4}{5} \times \frac{1}{2} \times \frac{4}{5} = \frac{1}{4} \text{ about.}$$

The stress or pull in the cable just outside the hawse pipe is evidently of the same nature as that of a pull in a tow rope (see Chapter XXVII.); and if

h = height through which the anchor is raised per revolution of the engine shaft, or the distance through which the ship is pulled—

$$\text{Stress in cable in } lb. = \frac{2\,P\,.\,L\,.\,A}{h} \times E \quad . \quad . \quad (1).$$

In the above formula for the same engine and gearing L, A, and h are constant qualities, and E is also approximately constant; and thus—

Stress in cable $\propto$ P (the pressure in the cylinders).

Or, in a simple case where the weight W of the anchor and cable is known, and the rate x (generally 25 feet per minute) of raising the anchor is specified—

$$\text{Effective H.P.} = \frac{\text{W lb.} \times x \text{ (feet per minute)}}{33{,}000},$$

which ,, $= \text{I.H.P.} \times E.$

$$\left.\begin{array}{l}\text{Stress in cable} \\ \text{in lb.}\end{array}\right\} = \frac{2\,P\,.\,L\,.\,A\,.\,N}{N \times h} \times E, \text{ from} \quad . \quad . \quad . \quad (1)$$

$$= \frac{\text{I.H.P.}}{N \times h} \times E \times 33{,}000, \text{ in which } N \times h = x \text{ evidently,}$$

$$= \frac{\text{I.H.P.} \times 33{,}000 \times E}{x}.$$

In actual practice the weight W includes an allowance for the holding force of the anchor when embedded in the sea bottom, and also an allowance for the frictional resistance of the cable up to the point of suspension to the cable holder. An overstress, or *strain*, is prevented by limiting the pressure in the engine cylinders, so that the power exerted by the engine shall not be in excess of that allowed for the cable stress.

Example.—Find the I.H.P. of engine required for lifting a weight of 35 tons equivalent to the cable stress) at the rate of 25 feet per minute.

$$\text{I.H.P.} = \frac{35 \times 2240 \times 25}{33{,}000 \times E} = \frac{60}{E} \text{ nearly.}$$

And if $E = \frac{1}{4}$, then I.H.P. of capstan engine

$= 240$ nearly, which is not uncommonly fitted in large battleships and cruisers.

It should be remembered that a capstan engine is fitted with a differential reversing valve and normal slide valves, and that the engine is almost entirely non-expansive working. The steam con-

sumption per I.H.P. is therefore very large, and consequently a large boiler power is necessary for working the capstan and weighing anchor ; and roughly, a boiler which will produce 1000 or more I.H.P. when supplying the main propelling engines, will barely supply a capstan or steering engine *working continuously* at 240 I.H.P.

Williams-Janney Variable Speed-Gear.—This gear is applicable to steering-gear, capstans, windlasses, hoists, cranes, winches, etc., which require considerable variation in speed in operation, and a large initial power at the instant of application. Power is supplied from an electric motor running at constant speed in one direction only, which is usually of the shunt wound type.

The variable speed-gear consists essentially of a pump, driven by the electric motor, with which it has a common shaft, and a hydraulic motor, which is of similar construction to the pump ; the hydraulic medium is oil, which provides for efficient lubrication of the internal parts of both the pump and the motor.

Fig. 389 shows the internal working parts of the pump barrel *C*, and the motor barrel *D*, separated by a fixed central valve plate *E*, which carries two roller bearings of which one is for the end of the pump spindle *A*, and the other for the end of the motor spindle *B*. When closed up the whole of the parts are enclosed in a common cylinder, and *C* and *D* are in working contact with the plate *E*. The socket ring *F* is permanently tilted, as shown, and housed in a casing *G*, with which it is in working contact through two sets of roller bearings, one of which takes the axial thrust and the other any radial thrust.

Similarly the socket ring *H* of the pump is housed in the casing *J*, which is mounted on trunnions *K*, and can be tilted as required by the rotation of the rod and vertical screw *L*, acting through trunnion nuts. The screw *L* has a coarse thread and requires less than two turns from the central position in either direction to give the full tilt allowed.

The whole interior of the casing is completely filled with oil through plug holes provided at the top of the fixed plate *E*. The pump spindle is rotated continuously by an electric motor or some other power at a constant speed, generally about 600 revolutions per minute, and the barrel *C* and the socket ring *H* are keyed to the shaft and revolve with it. *H* is mounted on a universal joint drive. The housing *J* does not revolve, and being fixed at any inclination by the screw arrangement *L*, it constrains *H* to move a corresponding oscillation during each revolution of the spindle, and thus moves the pistons, nine in number, to and fro in the cylinder barrel *C*, and pumps the oil under pressure through

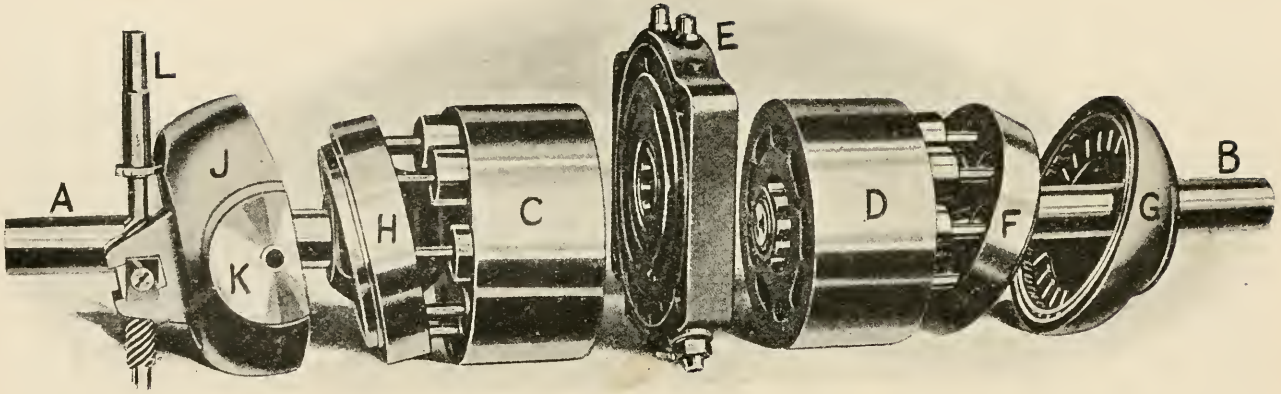


Fig. 389.

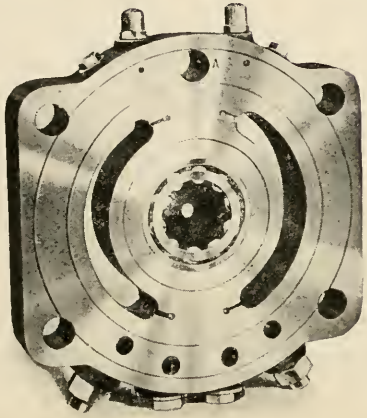


Fig. 389B.

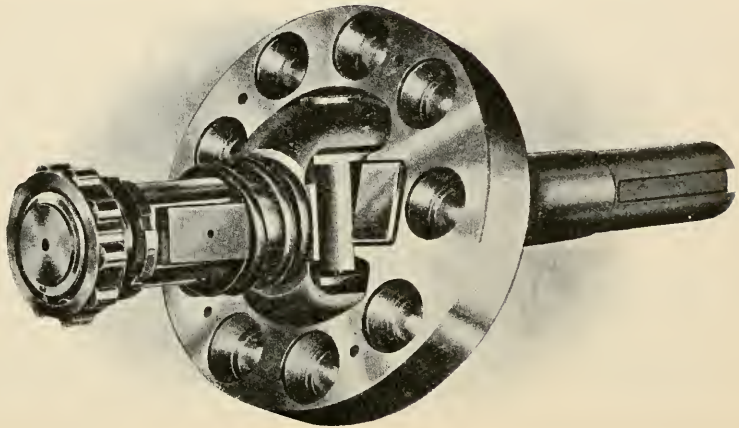


Fig. 389A.

Figs. 389, 389B, 389A.—Williams-Janney Variable Speed-Gear.

the ports in *E*, and the cylinder barrel *D*. The oil pressure operates the pistons socketed in *F*, and thus rotates the spindle *B*, on which *D* and *F* are mounted ; the socket ring has a permanent tilt or inclination.

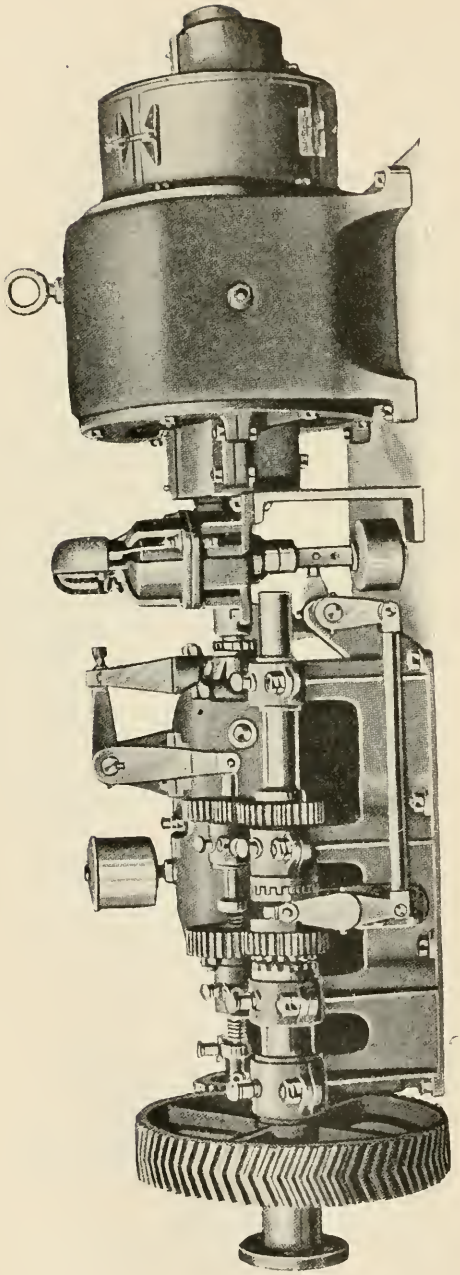
By tilting *J* at the opposite angle to that shown in the figure, the motor shaft *B* is driven in the opposite direction of the rotation, and at any point between the two extreme positions of tilt variation of speed is obtained between the pump spindle, driven at constant speed, and the motor spindle which corresponds to the tilt of *J* as effected by the rod and screw *L*. When the tilt has no inclination in either direction, and is, therefore, at right angles to the shaft axis, no pressure is produced by the pump, and no motion is communicated to the motor ; thus between the two extreme points of tilt an infinite variation of speed is obtained in either direction of rotation.

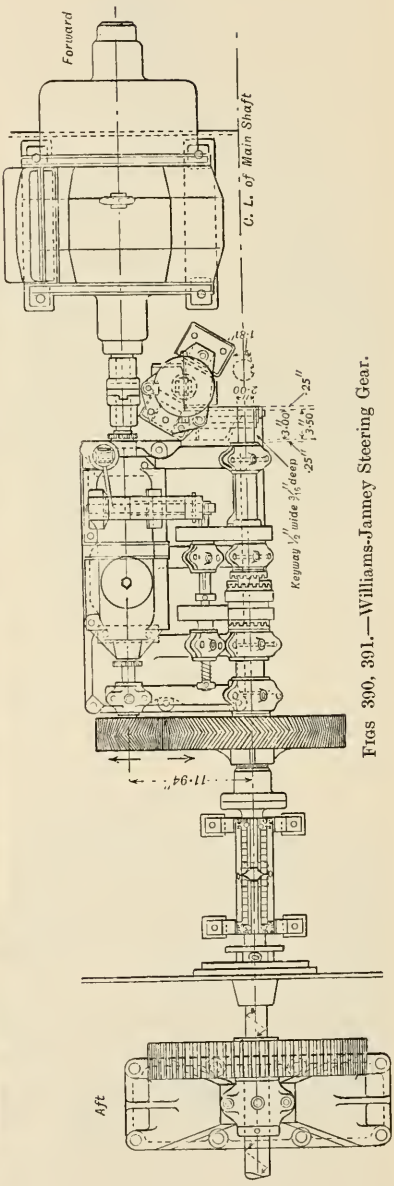
Fig. 389A shows details of the socket plate *F* and Fig. 389B details of the fixed central valve plate *E*. The axes of the nine cylinders in the cylinder barrel are parallel to the shaft axis, and the barrel is driven by two keys on the shaft, which are a sliding fit in the barrel keyways. The inner ends next the central valve plate are provided with oval ports, which as the barrel revolves are successively brought into communication with the two ports in the valve plate ; the difference in area between the ports and the cylinder keeps the faces of the cylinders in working contact, except for the oil film, with the central plate faces. When the oil is not under pressure, and therefore no transmission of power is taking place, a spring, bearing against a flange formed on the spindle, maintains contact between the working faces.

In each small cylinder formed in the cylinder barrel, works a grooved trunk piston fitted with a connecting rod having spherical ends, one butting on the piston and the other in the similar spherical-shaped hole in the socket plate. The socket plate is driven from the pump shaft by a universal joint, and the motor socket plate is of similar construction.

In the type above described the pump and motor are in a single casing, but in another type the pump and motor are in two separate casings, connected by two pipes corresponding to the two ports in the central valve plate. This latter system can be applied at a considerable distance between the pump and the motor, and is useful for transmission for various independent engines and machines.

The application of the variable speed-gear to a steering-gear of the right- and left-handed screw type is shown in Figs. 390 and 391. In





Figs 390, 391.—Williams-Janney Steering Gear.

this type the change-over from power to hand steering is operated automatically by means of a solenoid placed in series with the electric motor, so that, in the event of a failure of the current, the solenoid becomes de-energised and a falling weight attached to the plunger of the magnet carries the change-over clutch from the power to the hand-steering position, and the steering of the ship is continued from the same wheel and through the same shaft.

A steering-gear of the simplest type consists of a pump as above described, connected by pipes with rams used as motors and applied directly to the tiller through a trunnion crosshead bearing. This is probably the most efficient form of power steering-gear, and an efficiency of 80 to 85 per cent is claimed for it.

Boat Hoists.—The Williams-Janney variable speed-gear has been supplied to several battleships for boat-hoists. A weight of 18 tons can be lifted at the rate of 30 feet per minute, or 6 tons at 90 feet per minute. In other installations 16 tons are lifted at 80 feet per minute. No brake is required as the variable speed-gear acts as a positive brake.

CHAPTER XXX

WATER-TIGHT, PUMPING, AND FIRE SYSTEMS

Water-tight Subdivision.—A ship of any importance is divided into a number of *water-tight compartments*, so that in the event of one or more being flooded by grounding or collision the ship may still remain afloat. These several compartments are separated by steel-plate bulkheads, strengthened as necessary by vertical steel beams of Channel, *H*, or *Z* section. The principal bulkheads extend across the whole transverse width of the ship and are known as *transverse* bulkheads, and by their means the ship is divided into nine or ten or more *main* water-tight *compartments*. Longitudinal bulkheads extend between the transverse bulkheads as necessary, and thus subdivide the main compartments into a large number of small compartments which assist, if required, in keeping the ship afloat. The largest compartments are usually the boiler rooms (three or four or more in large battleships, cruisers, and passenger steamers) and the engine rooms (two or more). Next in cubic capacity are the coal bunkers, which are separated from the engine and boiler rooms by longitudinal bulkheads, and cargo holds, which may extend the whole transverse width between the inner outer bottoms. There are other large compartments, such as the capstan engine, submerged torpedo, steering, and tiller rooms, besides a great number of smaller spaces for stores and ammunition. Within the outer bottom, and 3 or 4 feet from it, there is another and inner bottom, and between these two inner and outer bottoms are a number of water-tight compartments, of which those near the keel are known colloquially as the *double bottoms*, and those near the water-line as the *wing compartments*. Thus, in case of collision, say, about amidships and in wake of the boiler rooms, three bulkheads must be pierced (viz. the outer and inner side or bottom plating, and the bunker bulkhead) before water gains access to the boiler room. By this system of subdivision a

well-built steel ship is well protected from sinking after collision or grounding.

Water-tight Doors.—For access to the various compartments and for communication between them it is necessary to have doors which, when closed, make each compartment a water-tight space. These doors are of various types according to their position and importance.

For the wing compartments, double bottoms, lower parts of the

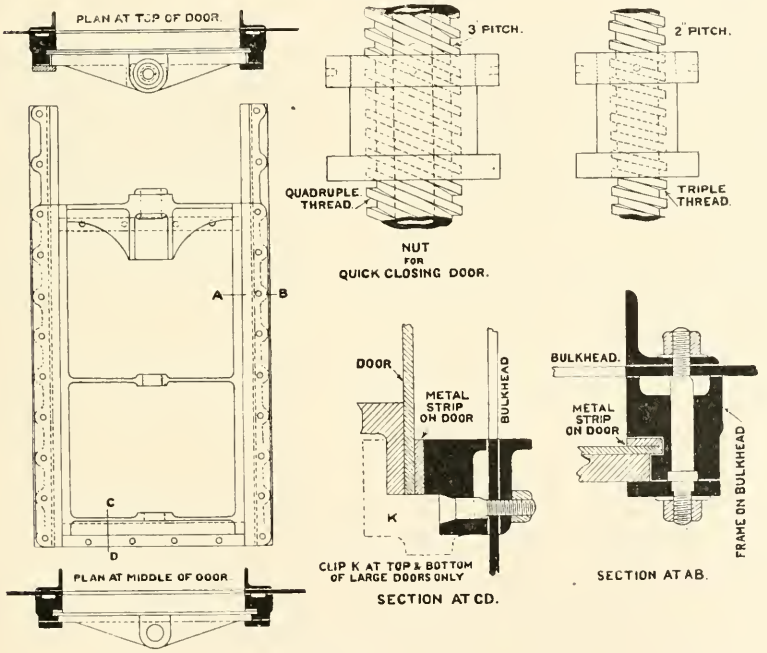


Fig. 392.—Vertical (Geared) Water-tight Door.

hold, forward and after (narrow) parts of the ship, and a few small store rooms, ordinary *manhole doors* (similar in character to those used on top of cylindrical boilers) secured by bolts and butterfly nuts are fitted as necessary, one or more to each compartment. These doors are only opened for cleaning, painting, or examination of the plating of the compartment, and are always closed and secured at sea.

For the main transverse and bunker bulkheads, where the height permits of their use, *vertical* (*geared*) *water-tight* doors are fitted, as

shown in Fig. 392. All geared doors are connected through rods and gearing with, and can thus be worked from, the main or upper deck, as well as from one or both sides of the bulkhead near the door. In recent warships the bulkheads between the several boiler and engine rooms are *solid*, *i.e.* no doors are fitted in them. Bunker doors are, however, necessary between the several bunkers and boiler rooms, so that the coal can be brought to the fires.

The vertical sliding water-tight door consists of a flat plate strengthened by ribs on the back (as shown in Fig. 392), and it is faced with a continuous strip of brass or gun metal where it comes in contact with the frame to make a water-tight joint. The frame is bolted to the bulkhead and makes a permanent water-tight joint with it; on each side there is a vertical wedge-shaped groove, tapered about $\frac{3}{16}$-inch per foot run, in which the door can be moved up or down. The door is raised and opened by means of a long screwed spindle working in a nut fixed in the top rib of the door. When the door is closed down, the wedge shape presses the metal face against the frame and completes the joint. It is necessary that these doors should be closed quickly, and, as shown in one of the small views, the spindle is sometimes made with a square three- or four-threaded screw; this quick-closing arrangement is correspondingly slow when opening, but this is a matter of no importance. The details of the ordinary construction are shown in the sketches.

Fig. 393 shows a vertical section and plan of a *deck plate and indicator* used in connection with the gearing for working a water-tight (geared) door. The cap *D* is first removed, and then a square socketed spanner is inserted and fitted over the end *E* of the gear spindle *F*. The rotation of the spindle raises or lowers a nut, *G*, which, being prevented by side lugs from rotating, rotates the indicator cylinder *H* by means of extensions of the lugs into nearly vertical grooves. On the top of *H*, and level with the deck, an arrow is engraved which points to the position of the door, whether open or shut.

In another arrangement, the indicator is operated through epicyclic gearing (Fig. 394, Gibson's Patent), which is probably more durable and less likely to jamb.

The *horizontal sliding water-tight door* is of similar construction to the vertical variety, but, instead of a long screw, a rack-and-pinion gearing is used for opening and closing it, as shown in Fig. 395. The weight of the door is supported by rollers *R*, and a space is generally left below the lower wedge-shaped groove to allow the dirt to fall

clear and thus not block the passage of the door when closing it. Over the sill of the doorway a sill-plate is fitted which is pushed away automatically by the door when closing it. At the back edge of the door, clips are fitted which swing automatically into place and assist in holding the door closely against the frame.

Hinged water-tight doors (Fig. 396) are used in places of minor importance, and above the water-line particularly. The frame is riveted to the bulkhead, and a strip of india-rubber or sheet asbestos

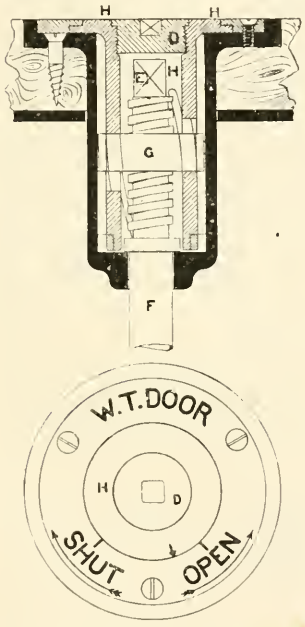


Fig. 393.—Deck Plate and Indicator.

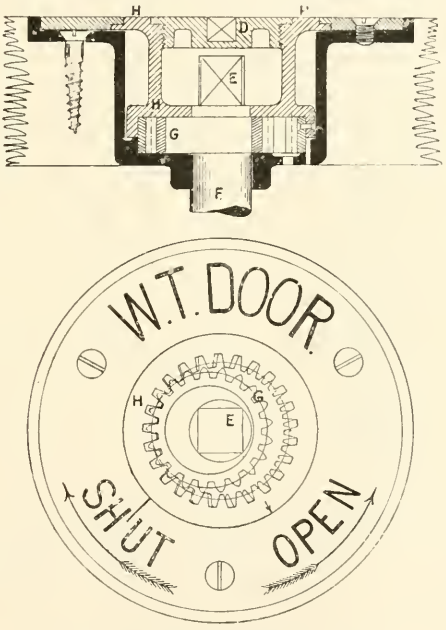


Fig. 394.—Epicyclic Geared Indicator.

is fitted to the frame and forms a continuous face, against which one edge of an angle steel (riveted to the edge of the door) is pressed when the door is closed and clipped up. These clips are a weak and unsatisfactory fitting; they are liable to fracture under a sudden stress, and are usually incapable of maintaining water-tightness under a heavy pressure.

Below the water-line and over important deck openings geared sliding hatches, of similar construction to the horizontal (geared) door and to those used between the upper and lower bunkers, are fitted; but for deck openings above the water-line, especially in the

armour decks, *armoured hatches* are fitted and hinged to the deck. These hatches are frequently very heavy, and differential chain tackles are used for opening or closing them. These hatches are not usually an

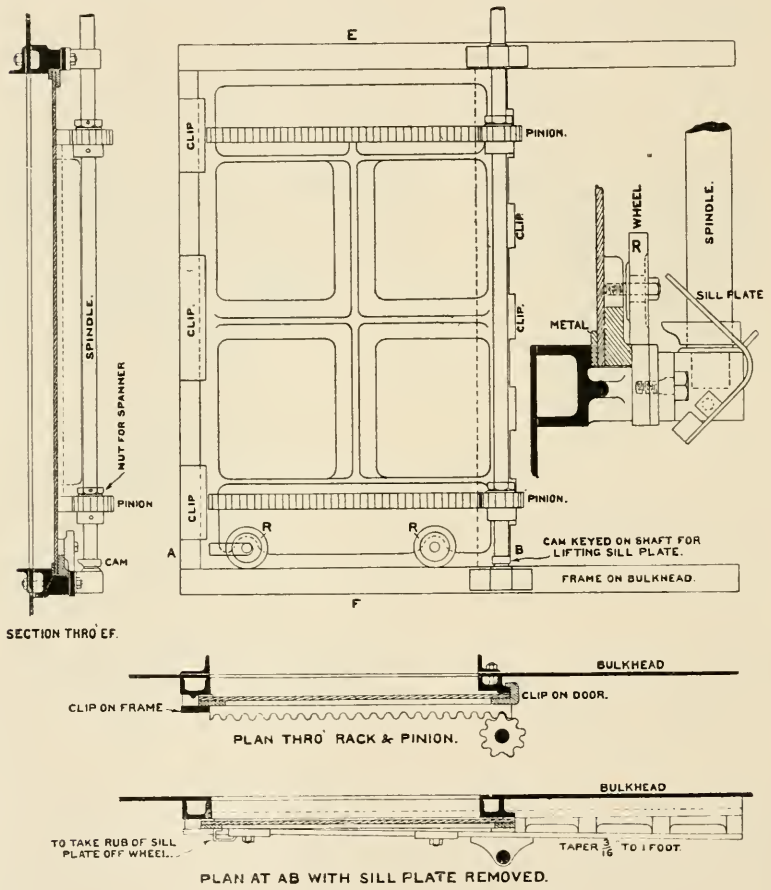


Fig. 395.—Horizontal (Geared) Water-tight Door.

entirely water-tight fitting; but clips are fitted, and generally a good joint can be secured by pressing them down on soft packing or gasket.

For the purposes of drainage from one compartment to another *sluice valves* are sometimes fitted instead of ordinary stop valves. The sluice valve is a vertical (geared) door of small size (from 3 inches square upwards), and in important positions is generally fitted with

an indicator plate so as to be workable from the main or upper deck. Under certain circumstances of grounding or collision sluice valves become a dangerous fitting, and in one or two well-known instances disaster has been only narrowly averted.

In all recent ships, both mercantile and naval, each main transverse compartment is usually self-contained in regard to ventilation, and ventilation trunks are independent of those in other main compartments. This arrangement not only increases the safety of the ship but also improves the ventilation. Since attention was first called to these important matters of water-tight fittings and ventilation, in the first edition of this work in 1896, considerable improvement has been made; and there only remains to add the caution of not relying on any pumping arrangement, however efficient it may be, but rather to ensure every bulkhead being really water-tight and efficient enough to keep the ship afloat.

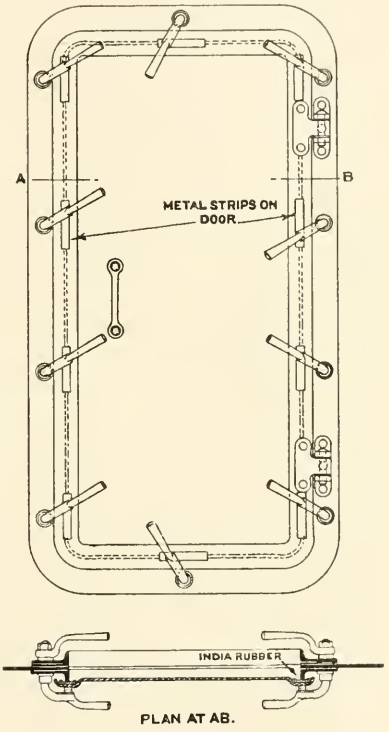


FIG. 396.—Hinge-clip Water-tight Door.

Stone-Lloyd Water-tight Door Control System. — In this arrangement the doors are operated by hydraulic power supplied by pumps always at work. The hydraulic pressure pipes, as fitted in the *Mauretania*, are in connection with a screw-down valve, the operation of which from a pedestal on the bridge, by means of steel wires and chains, admits pressure into a subsidiary line of piping and closes all the valves simultaneously. When the clutch-gear for closing the doors is moved to admit pressure, a large gong is rung at each bulkhead door station and warns every one of the impending closing of communication. A special arrangement is made so that a man shut into any compartment can liberate himself through the door, but the operating lever must be held continuously or the door recloses

automatically. The door closes in all cases slowly, and the power is sufficient to break up any obstructions.

Pumping Capacity Available.—In a large warship the following engines are available for pumping water from the ship's bilges :

In each engine room—

2 Main circulating engines driving centrifugal pumps, each with a capacity of about 5000 tons per hour when running at 400 revolutions per minute, against a head of 36 feet.

2 Fire and Bilge pumps, each with a capacity of about 50 tons per hour.

In each boiler room—

1 Fire and Bilge pump, with a capacity of about 50 tons per hour ;

1 Turbo-driven Salvage pump, with a capacity of about 1000 tons per hour.

In various parts of the ship—

4 to 10 Hand pumps, each of low capacity, but usable for putting out local fires in emergency and for pumping out small quantities of water from the bilges.

In addition, and particularly in small vessels, Bilge Ejectors of capacities up to 200 tons per hour are fitted in the main compartments.

In merchant vessels there are usually two fire and bilge pumps in each main engine room, and also a salvage pump of fairly large capacity, which is usually known as the *ballast* pump and used for shifting ballast water to alter the trim of the vessel.

During the war great additions have been made to the pumping power of all vessels, and this has taken the form of salvage pumps, frequently driven by small turbines, which will now be described.

Allen Pipeless Salvage Pump.—More than fifty years ago, H.M.S. *Inflexible*, the strongest battleship of her day and for some years later, was fitted with centrifugal circulating pumps for the main condensers, which in emergency could be used to pump from the bilges as in other vessels, but these were arranged on vertical shafts direct driven by engines fitted on the engine room bulkhead above the water-line and protected by the vertical armour outside the ship from shot and shell. Incidentally it may be mentioned that the armour was a foot thick, and there were two thicknesses, making up 2 feet of armour with a considerable backing of wood. Friedmann ejectors of large capacity were fitted to the other main compartments.

As shown in Fig. 397, the Allen arrangement has a vertical shaft with a small impulse turbine at the top end and a centrifugal pump

at the lower end just above the inner bottom. The turbine has a single stage impulse wheel designed for a working pressure of 150 lb. per square inch, exhausting alternatively either to the atmosphere or to the closed exhaust against a back pressure of 25 lb. per square inch. The fittings are similar to those illustrated and described in connection with small turbines generally, but the shaft being vertical its weight

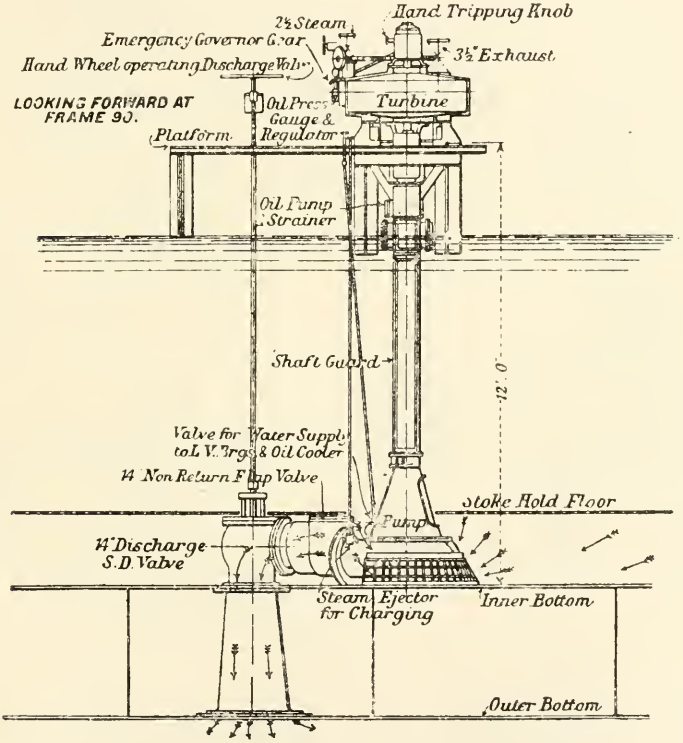


FIG. 397.—Allen Pipeless Salvage Pump.

and that of the turbine are taken on a Michell thrust and another similar thrust takes the weight of the lower part of the shaft and the impeller of the pump ; the two parts of the shaft are coupled by a claw flexible clutch. The normal speed is 940 revolutions per minute when delivering 1000 tons of water per hour against a head of 35 feet.

A sectional elevation of the pump and valve arrangement is shown in Fig. 398. The pump is constructed entirely of gun metal and the

guide bearings are lined with lignum vitæ. The pump is self-charging and runs satisfactorily when submerged or flooded until the water reaches the steam turbine. No suction pipe is fitted and only a short connection is necessary to convey the delivered water to the sea through a non-return valve and a stop valve operated from alongside the turbine.

B.B.C. Combined Fire and Salvage Pump.—This combination is constructed by Brown-Boveri & Co. and direct driven by one of their small turbines of the impulse type with one impulse and several velocity compounded stages on a single wheel, as shown in Fig. 399. When

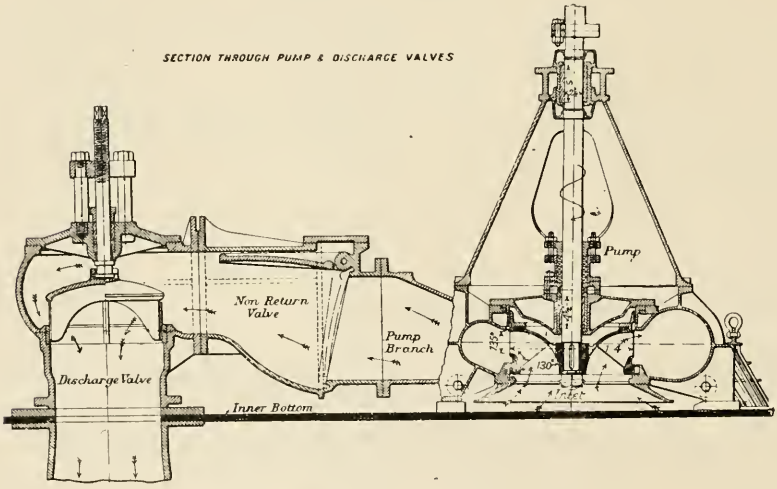


Fig. 398.—Allen Turbo Salvage Pump. (Section.)

used as a fire pump the revolutions per minute are 2800 and water delivered about 192 tons per hour against a head of 100 metres or 140 lb. per square inch. When used as a salvage pump, the revolutions per minute are 1700 and the water delivered about 1750 tons per hour against a head of about 10 metres or one atmosphere above atmospheric pressure. The total weight of the pump set is 5290 lb. or about 47¼ cwts.

The speed of the turbine is regulated by throttling the steam through the valve gear shown in Fig. 400. The governor-gear is controlled by the water under pressure, which ensures a constant discharge pressure of the fire pump and acts only when this pump is in service. When the salvage pump is in use regulation is effected by hand through

the main stop valve, and the governor remains full open by means of its spring control.

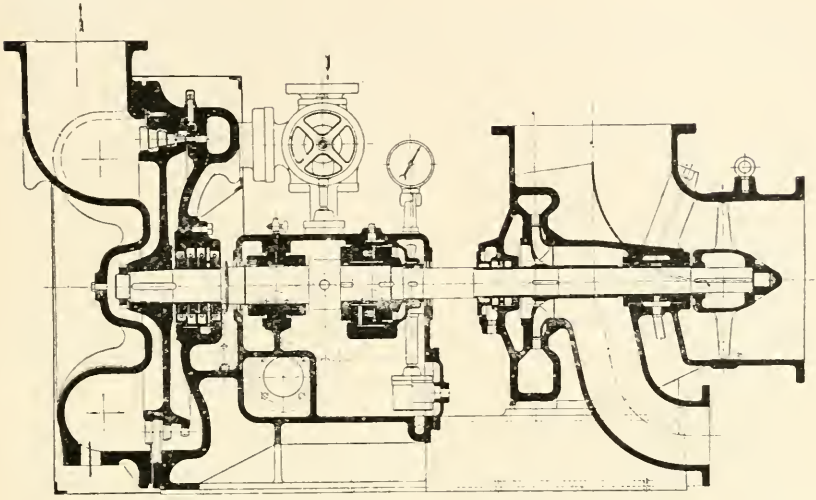


FIG. 399.—Brown-Boveri Combined Fire and Salvage Pump.

The single stage fire pump is a high pressure centrifugal pump fitted as shown in Fig. 400, with fixed outlet guide blades, of which

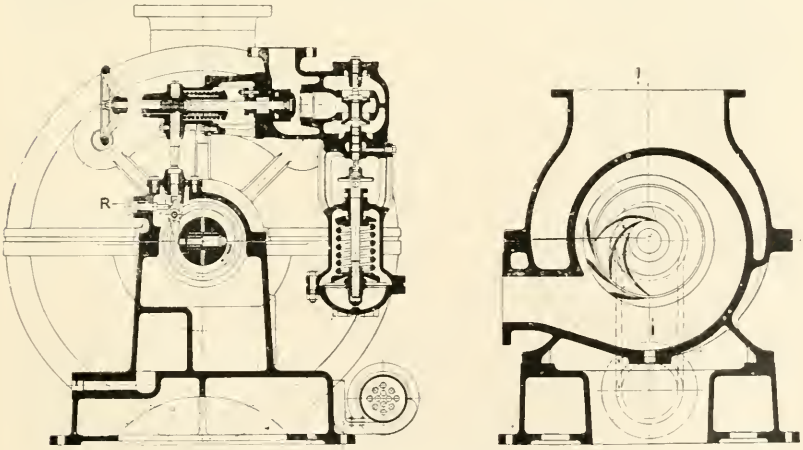


FIG. 400.—Brown-Boveri Turbo-pump Governor Gear.

only a few are shown. Axial pressure on the pump impeller is provided for sa in the firm's usual practice.

The salvage pump is of the helocoidal type with axial helices.

The two pumps work independently of each other, and are separated by a partition which also serves as a support for the bearing between the two impellers. This bearing is lined with antifriction metal and is water lubricated by the pump in service. The parts of the shaft in contact with the water are covered with a nickel steel sleeve to prevent rusting. The lubrication water circulates through a coil fitted in the oil reservoir in the turbine casing, and all impurities, such as seaweed, etc., which would cause undue wear of the bearing are excluded or deposited on the way because of the slow circulation through the coil.

The general arrangement provides for easy access to all the working parts, and the set is entirely independent of all the other auxiliaries ; and is provided with its own forced lubrication oil pump, which is of the gear-wheel type driven by a worm and worm-wheel from the turbine shaft.

Fire and Bilge Pumps.—Fig. 401 illustrates one of these pumps as made by Messrs. Mumford & Co. for fire and bilge, feed, and ballast pumps. For feed and high pressures the pump valves are usually of the group type, and for fire and moderate pressures of the spring-loaded type, as shown in the figure. The pump plungers are fitted with ebonite packing rings for high pressures, or they can be fitted with a renewable solid uncut metal packing ring with water grooves for low and moderate pressures. There are the usual two sets of valve chests for each pump barrel, one for the up-stroke, as shown, and another for the down-stroke, not shown in section.

The main slide valve is of the ordinary D type and works on a flat cylinder face with three ports. By means of a forked sleeve piece, which can be put on or removed by removing the slide chest cover, the slide valve is operated by a shuttle valve working in small cylinders at each end of the valve chest. The shuttle valve, and by its means the slide valve, is operated positively through the medium of the valve spindle worked by levers attached to the piston-rod crosshead.

In the position shown, the bottom end of the shuttle valve is open to steam through a small port in the valve and a corresponding port in the valve chest communicating with the bottom end of the cylinder, and consequently boiler steam. The top end of the shuttle is in communication with the exhaust through similar ports at the top end of the cylinder. The main slide valve has been pushed upwards by the movement of the shuttle valve in consequence of the difference of

pressure below and above it, and thus admits steam below the piston and drives it upwards. At a certain predetermined point of the piston stroke the shuttle valve is pulled downwards by the lever action, steam is admitted above it and exhaust opened below it, and its downward travel is completed independent of the slide rod, which is given

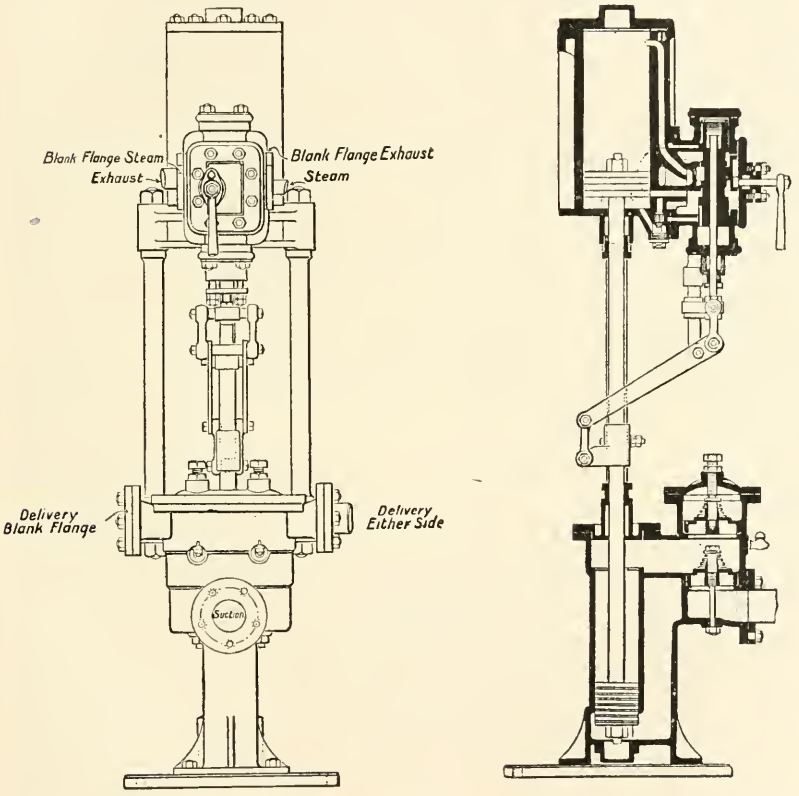


FIG. 401.—Mumford Fire and Bilge Pump.

sufficient up and down play for this purpose. In the same time the piston completes its upward stroke and starts downward under the influence of the pressure admitted by the slide valve.

All the working parts can be easily examined and removed, and by taking off the top and back covers of the slide valve chest, both the shuttle and main slide valves can be removed and examined without disconnecting any pipes. The various pipe connections can be fitted

either right or left of the pump as convenient, and in ordinary practice two pumps are frequently fitted side by side with facility for either separate or combined service as may be required.

Each pump is driven by a separate engine. They are usually of the same size, with interchangeable working parts, and fulfil the same purpose. It is usual to keep the fire engines only for use in pumping on deck, so that only clean water shall pass through them, but in cases of urgency they are used as bilge pumps. In recent ships the steam and water pistons are fitted to the same piston rod, and no gearing or crank shaft is used. Independent suction pipes are led to each pump from each of the machinery and boiler spaces, from the sea, and from the main drainage suction system, in connection with the various compartments of the ship. Each pump can deliver either into the fire-main or overboard. The pump valves are frequently made of vulcanised india-rubber; but sometimes flat metallic valves of the Kinghorn type are used, and are generally more reliable.

Bilge Ejector.—For small vessels the usual fire and bilge engines are too heavy, and can only be connected with the various compartments by long and comparatively heavy pipes, and therefore a bilge ejector is generally fitted to each compartment. About ten to fifteen such ejectors are fitted in a torpedo-boat destroyer, and each is capable of ejecting about 60 to 100 tons of water from the bilge per hour.

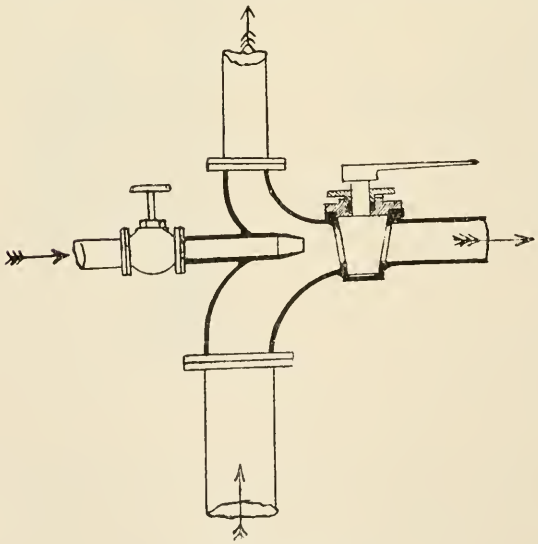


Fig. 402.—Bilge Ejector.

A bilge ejector is shown in Fig. 402. The top pipe is connected with the hand pump, and must be shut off when the ejector is working. The bottom pipe is led down to the lowest part of the bilge, and the end is left open and covered by a strainer. Steam is admitted through

the small pipe, valve, and nozzle shown, and the mixed steam and water is blown overboard through the cock and pipe to the right. If the water is not immediately drawn up when the discharge cock is opened and a little steam turned on, the apparatus can generally be started by keeping the cock closed and turning on steam gently. Condensation in the suction pipe (increasing the nozzle energy, see Chap. XX.) draws up some water, and a sudden opening of the discharge cock generally sets up a discharge, when the steam valve can be opened gradually until a full discharge is obtained. Within the limits of effectual operation, the ejector should be fitted as close to the bottom of the bilge as possible, and as no suction is then exerted, the certainty of action is increased ; but it is necessary to keep the ejector at a fair height above the bottom of the bilge for easy manipulation when the bilge is flooded, and therefore they are generally fitted above the water-line. The efficiency of the ejector depends very largely on the internal shape of the nozzle.

The objection to the use of ejectors is the loss of fresh feed water, which has to be made up, because it cannot be returned to the boiler. There is, however, no mechanism to get out of order ; the dirt and small ashes are passed overboard without stopping its working, and the space and weight required are very small.

In some of the larger ships the ejector is fitted with a series of nozzles, and in the bilge without a suction pipe. The steam valve can be worked by gearing, from some point well above the bilge. The momentum of the steam will frequently raise the water 30 feet or more, but the consumption of steam is very great, and, of course, the fresh water is wasted.

Latrine Pump.—A small engine, of the same type as the fire engine, is sometimes fitted in larger ships for pumping through the fire-main and flushing the latrines. It is not usually available for any other purpose.

Quantity of Water entering a Vessel through a Hole.—For this calculation the following figures are useful :

1 ton of sea-water occupies 35 cubic feet.

1 cubic foot of sea-water weighs 64 lb.

1 gallon of fresh water weighs 10 lb., and sea-water weighs about 3 per cent more.

1 ton of fresh water contains 224 gallons.

The velocity with which water will enter a ship depends on the *head*, or depth below the water-line. Call this H ; then—

$$\text{The velocity} = \sqrt{2 \cdot g \cdot H}$$

where $g = 32$, approximately.

$$\text{Therefore velocity} = 8 \cdot \sqrt{H} \text{ in feet per second.}$$

From which, if A = area of hole in square feet,

$$\text{Tons entering per hour} = 820 \cdot A \cdot \sqrt{H} \text{ about.}$$

By simple calculation, the quantities in tons which will enter a ship through a hole of 1 square foot in area are as follows :—

Depth of Hole below Water-line.	Tons of Water entering per Hour.	Remarks.
1 foot	820	For each 1 sq. ft. of area of hole.
4 feet	1640	
9 ,,	2460	
16 ,,	3280	
25 ,,	4100	

By comparison with the pumping power, it is evident that only the leakage through a very moderate-sized hole can be kept under, and that the means of keeping a ship afloat depends principally on the water-tightness of the various compartments. It is hardly necessary to emphasise the immediate necessity of localising any leakage and isolating the compartment affected.

Fire Arrangements.—The fire and bilge pumps are available for the purpose of putting out a fire, and in addition, each hand pump can be connected with the common delivery pipe, known as the *fire-main*. The fire-main run of piping is situated below the armoured deck or water-line, and runs fore and aft nearly the whole length of the ship. There are branches with valves and fittings for hose connections in the various compartments, and in each main transverse water-tight compartment a branch, known as the *rising-main*, leads up to the compartments above the armoured deck and water-line. A valve is fitted close to the fire-main, so that each rising-main can be shut off in case of injury to it, and at certain bulkheads shut-off valves are fitted to allow various sections of the main run of pipe to be shut off and repaired. The forward rising-main is commonly used for washing down the cable when getting up anchor, and is termed No. 1 rising-main, and the remainder are numbered consecutively—No. 2, No. 3, etc.

In warships, valves and cocks connected directly with the sea are

fitted for *flooding the magazines*, etc., when required. The flooding valve for each individual magazine is locked, so that it cannot be tampered with. One main sea connection is generally provided for each group of magazines, and branches are led from it, as necessary, to the locked flood valves.

The sea connections throughout the ship are now generally ordinary stop valves, made entirely of special brass. A short pipe,where necessary, extends between the valve (which is next the inner skin) and the outer bottom. On the outside around each opening a *zinc protector* is fitted. The zinc does not prevent galvanic action, but is more susceptible than either the steel bottom plating or the brass valve, and attracts to itself nearly all the corrosive and galvanic action. These zinc protectors are usually renewed on occasions of docking, or at least once every year. Similarly, protection from corrosion and galvanic action is fitted in the region of the stern bearings and screw propellers.

CHAPTER XXXI

VENTILATION AND FORCED DRAUGHT FANS, AIR-COMPRESSING AND HYDRAULIC MACHINERY

Air-compressing Machinery

In the Royal Navy, compressed air is used for three principal purposes :—

1. For supplying air to the fires in the boilers ; working pressure usually not exceeding 6 inches on the water barometer.

2. For removing the soot and small scoriae from the outer surfaces of the tubes of boilers and superheaters of the Babcock and Wilcox and other types, where manual appliances cannot be quickly applied ; working pressure usually not exceeding 50 lb. per sq. inch.

3. As the motive energy of torpedoes of the Whitehead type ; working pressure up to 2250 or 2500 lb. per sq. inch. And for forcing the torpedo from the directing tube (or gun) ; working pressure up to 750 lb. per sq. inch supplied from a reservoir.

Ventilation and Forced Draught Fans.—For ventilation the fans are commonly of the Sirocco type driven by electric motors which are light and occupy small space. In all ventilation schemes, whether by natural or artificial means, the first consideration should be based on the natural laws of convection currents enunciated and illustrated on page 32. If pipes or trunks are used in the system, they should assist the natural flow upwards or downwards with an absence of sharp bends or changes of direction which not only check the flow but reduce the efficiency and consequently increase the driving power necessary. Generally, exhaust ventilation is more effective and less local in its action than a plenum or forced supply, and exhaust usually admits of very short pipe connections compared with the plenum supply system.

For forced draught a centrifugal fan is nearly always used and

where a pressure is required to be maintained, as usually the case, the centrifugal fan is more efficient than a helical impeller ; but a helical impeller is more efficient for ventilation and simple circulation of air in crowded or warm spaces.

As noted in Part III., two principal systems of forced draught are in general use : the closed stokehold system for warships and the open stokehold with closed ash-pits for mercantile vessels ; also a large number of foreign warships are fitted on the closed ash-pit system in conjunction with the Kermode system of oil-fuel burning. In any system it is necessary to efficiency to obtain clear runs of ducts to and from the fans.

In warships the F.D. fans are usually slow running, 450 revolutions per minute, when maintaining a designed air pressure equal to a head of $4\frac{1}{2}$ inches of water ; in this case they are driven by single-cylinder reciprocating engines, which call for no particular remarks. In some cases, however, small turbines are used and run at from 1000 to 1500 revolutions per minute.

The capacity of a F.D., or other fan is measured by the volume of air which it will supply under a designed pressure, which for British naval purposes is generally $4\frac{1}{2}$ inches for oil-fuel burning and somewhat less for coal firing. On service these pressures may frequently be exceeded, especially when coal fires get dirty and clogged with clinker.

Considerable trouble has been experienced with F.D. fans owing to imperfect design and insufficient allowance for high peripheral speeds of the impeller, which requires careful dynamic balancing : in some few instances it has been found necessary to recast the design and rebuild the impellers.

Fig. 403 shows a general arrangement of a pair of turbo-driven fans, constructed by Brown-Boveri & Co., with a capacity at 1000 revolutions per minute of 25,000 cubic feet of air delivered under a pressure of $1\frac{3}{4}$ inches of water, and at 1450 revolutions per minute of 120,500 cubic feet of air under a pressure of 8 inches.

Fig. 404 shows the fans differently arranged for about the same capacity. The driving steam turbine is of the usual B.B.C. type, and its shaft is rigidly coupled to that of the fan so that two bearings only are required for the whole set. The bearing between the turbine and fan is a combined supporting and thrust bearing. Only one stuffing box is required and it is packed with carbon packing. Forced lubrication is fitted throughout, including a naturally cooled coil with

radiating fins, and is placed either in the suction or exhaust duct (engine room) in a current of air.

The steam supply for varying loads and speeds is controlled by a cylindrical slide valve, operated from the working platform, which admits steam through separate pipes to one or more nozzles as necessary. The fan impeller itself consists of wheels in parallel whose flanges and vanes are made of best Siemens-Martin steel. The volutes, or non-moving exterior parts next the fan, are made to suit the special requirements of individual ships. The capacity and number of fans fitted in

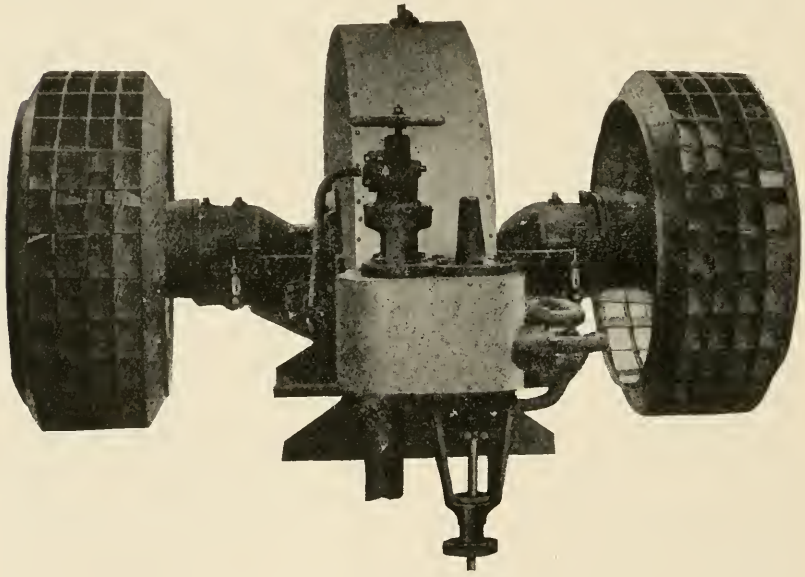


Fig. 403.—Brown-Boveri Turbo-driven Fans.

any vessel must necessarily meet the requirements of combustion as stated in Part III., which for an oil-fired ship would be about 9 cubic feet of air per minute at $4\frac{1}{2}$ inches pressure per 1 S.H.P. Thus in one of the largest new cruisers of 180,000 S.H.P. nearly two million cubic feet of air could be supplied per minute to the fires at full power ; some idea of the volume thus to be supplied may be gathered from an air-ship of similar capacity. The capacity of the F.D. fans in any vessel is generally much in excess of the volume required, so that a large reserve is available in case of emergency ; the usual design allows for one half the fans being unavailable.

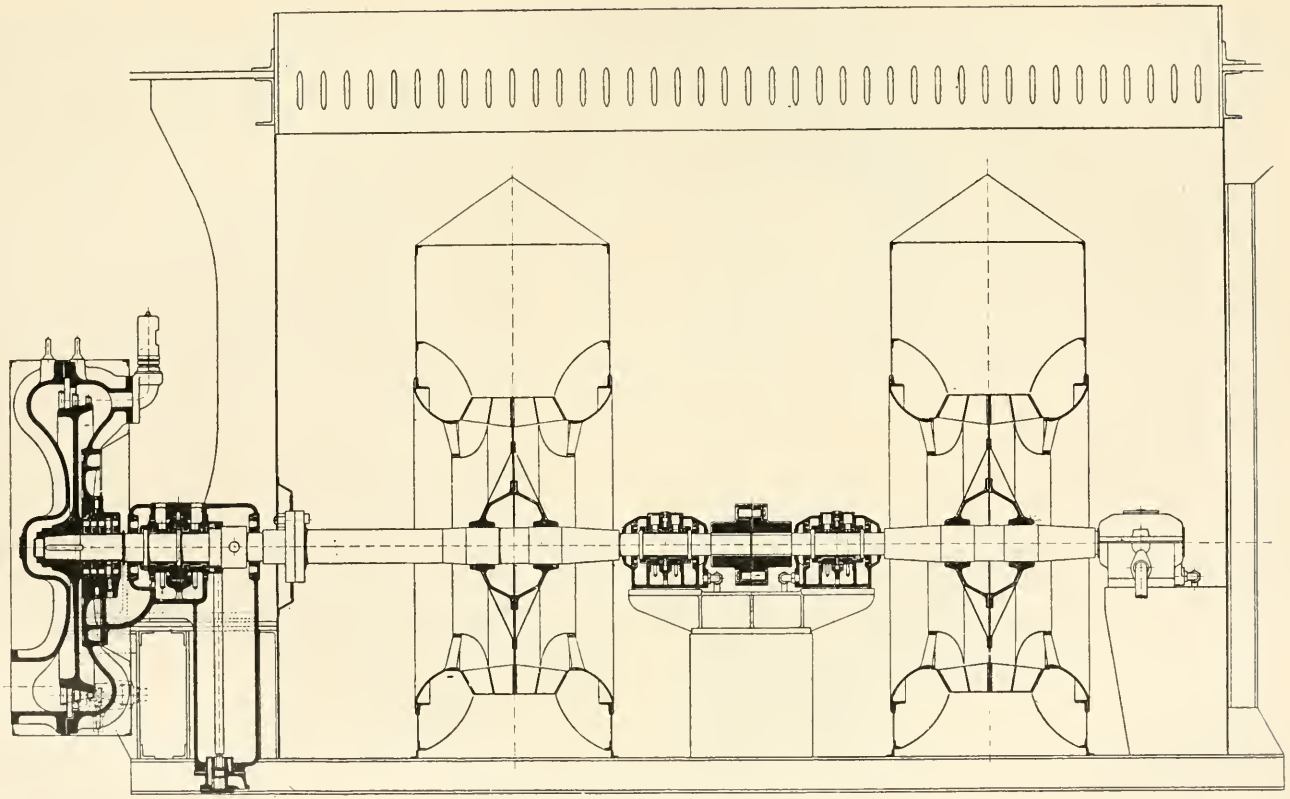


Fig. 404.—Brown-Boveri Turbo-driven F.D. Fans.

Blower for Sweeping Tubes.—Fig. 405 shows a Brotherhood compressor used for moderate air pressures of 80 to 250 lb. per square inch. The steam cylinder *A* is at the top, and the crank shaft and closed chamber *B* at the bottom, with the compressor cylinder *C* between. The crank shaft serves to drive the slide valve for controlling the supply and exhaust of the steam to the cylinder *A*, and also to limit the piston stroke and ensure its completion. The crank chamber is closed, and the crank and connecting-rod bearings are lubricated by oil compressed and forced through them by the oil pump shown. This pump is of the plunger type, without valves, and of similar construction to that used for forced lubrication of the electric light engine shown in Fig. 412. A small pump is also fitted, in connection with the slide-valve rod, to circulate water through the compressor cylinder jacket; by this means the air is delivered at nearly atmospheric temperature. The inlet water delivered by the pump generally enters the water-jacket space *D* at the bottom end of the compressor cylinder *C*; then passes upwards through the barrel jacket *E* into the top-end jacket *F* and thence to the outlet. Air from the surrounding atmosphere enters through the air inlet and inlet valves; the compressor is a double-acting pump; and the compressed air passes away through the air discharge. The inlet and outlet valves are generally made of thin gun metal and have a limited lift; they can be examined and renewed without removing any part of the remaining fittings, and the arrangement is a very convenient and easily accessible one.

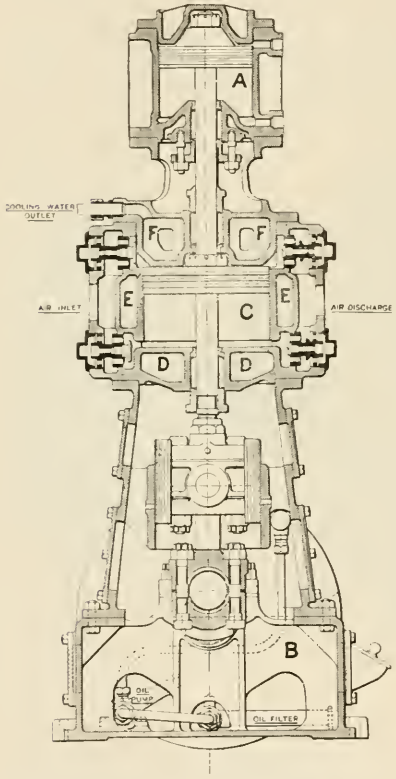


Fig. 405.—Air Compressor, Single-stage Type.

Multiple-stage Air Compressors.—As already explained in Chapter XV., it is necessary when compressing air to high pressures

not only to water-cool the compressor cylinders, but also to cool the air between the various stages of compression; otherwise the energy expended in doing the work of compression would be unduly great and

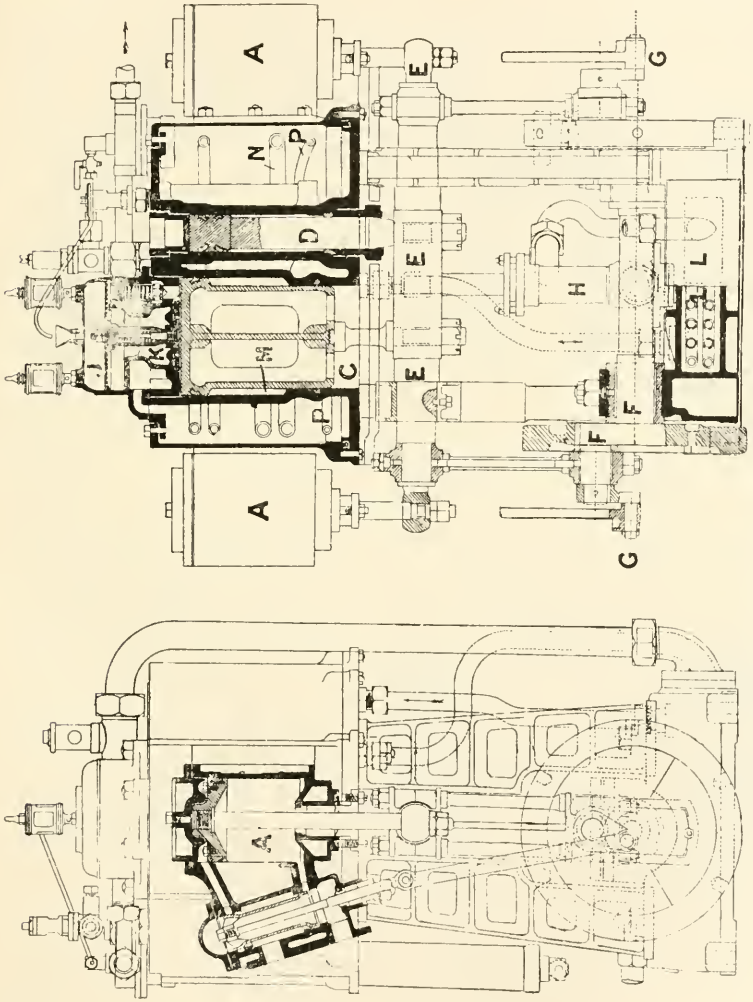


FIG. 406.—Brotherhood Four-stage Air Compressor.

the efficiency unnecessarily low. Usually three or four stages of compression are embodied in the apparatus; and the higher the pressure attained the smaller the diameter or stroke-volume of the compression cylinder.

In *Brotherhood's four-stage air compressor* (Fig. 406), there are two

steam cylinders AA and two compression cylinders C and D side by side; the pistons and plungers are all attached to a single cross-

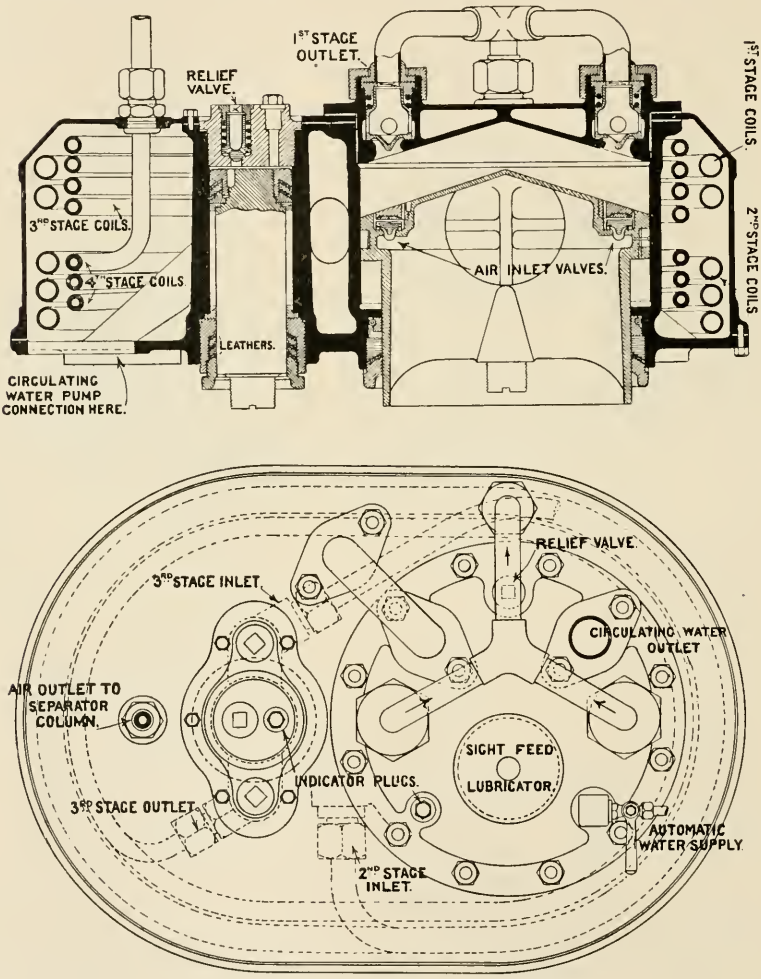


Fig. 407.—Belliss Four-stage Air Compressor.

beam E; and E is connected by double connecting rods with a crank shaft F, which in its turn operates the steam slide valves through small cranks G and G. A water circulating pump H of the plunger type is also attached to the crossbeam, and is worked through its

means. The steam cylinders are of the usual kind and are fitted with cylindrical slide valves, one of which is shown in section in the left-hand figure.

Each compression cylinder is utilised for two stages of compression; the lower pressure above and the higher pressure below the pistons respectively. Air from the atmosphere, with a little distilled water and sperm oil continuously supplied, enters through *J* and the inlet valve *K* above the left hand (and larger) plunger on the down-stroke, and fills the cylinder. On the up-stroke the valve closes, and the air is compressed and passes through a non-return valve and the first-stage intercooler coils *L* into the space *M* below the piston. On the next down-stroke the air is still further compressed, passes through a non-return and the second-stage intercooler coils *N* into the space *above* the smaller right-hand compressor piston.

In the second compressor cylinder similar operations are continued, and finally the compressed air, now at its highest pressure, passes through the fourth-stage cooler *P* into a separator. This separator drains the water away automatically, or it can be drained by hand, and the air is then discharged into a reservoir or torpedo as required. With very high pressures the removal of the water is attended with considerable difficulty, because the density becomes very great.

The circulating cooling water passes first through the casing containing the first-stage intercooler, and thence through the space surrounding the compression cylinders and containing the second, third, and fourth cooling coils, to the discharge outlet.

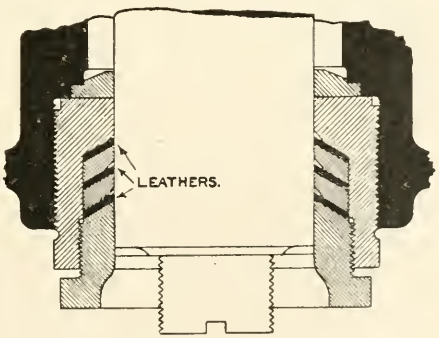


FIG. 407A.—Belliss Packing Leathers.

Another type of four-stage compressor, made by Belliss and Morcom, is shown in Fig. 407, and a further description is hardly necessary. The various connections are clearly shown, and the working stages can be easily followed. The Belliss method of fitting the leathers is shown in Fig. 407A, which also shows an enlargement of the packing gland and part of the small plunger rod of the H.P. stage of Fig. 407.

Hydraulic Machinery

Hydraulic (water) pressure is used in warships principally for loading and training heavy guns and turrets, for transporting and hoisting ammunition and projectiles to the turrets and guns, and for hoisting and lowering heavy boats and weights. In the mercantile marine it is sometimes used for loading and unloading cargo. In a few cases it has been applied for steering the ship, but no economical advantage is gained by its adoption in any of the mechanisms used for this purpose.

Until recently the engineer officer was entirely responsible for all hydraulic motors; but although he still deals with all important defects that may arise, and all repairs are naturally effected by the engineering staff both on board ship and in the dockyards, the Admiralty (very wisely, in the author's opinion) have to a large extent relieved him of the general routine duties, which are now carried out by the gunnery officer, who applies the motors to his special requirements. The motive power, being steam-driven, is of course supplied by the engineering staff, who are directly responsible for all hydraulic motors, such as boat hoists, which do not pertain to the guns. The Gunnery and Hydraulic Manuals deal very fully with all gunnery details, and consequently it is only necessary to herein describe those parts for which the engineer is directly responsible.

Hydraulic Pumping (Power) Engine.—The hydraulic motors derive their power from hydraulic pressure produced by one or more steam-driven pumps. The engine is usually a two-stage compound-tandem steam engine with the H.P. cylinder in front of the L.P. cylinder. The steam and pump cylinders are generally horizontally arranged, and the H.P. piston rod is directly coupled to a crosshead, to which the twin piston rods of the L.P. piston are also attached. The pump rod is also attached to this crosshead and is in the same axial line with the H.P. piston rod. Two such engines as above described are fitted side by side and coupled to the same shaft, with cranks at right angles, by the usual connecting rods which connect the crosshead and crank. The steam engine requires no special description, and is easily understood when seen.

One of the pair of pumps is shown in section in Fig. 408, and the essential point in its construction is that the sectional area of the pump rod is made exactly one-half that of the plunger. The plunger is fitted with two U- or L-shaped cupped leathers (Fig. 408A), and is entirely water-tight for one direction of motion only—from right to left, as

shown in the figure. The action is as follows: during the stroke, from right to left, water is drawn in through the suction valve and fills the space behind the plunger and below the intermediate valve;

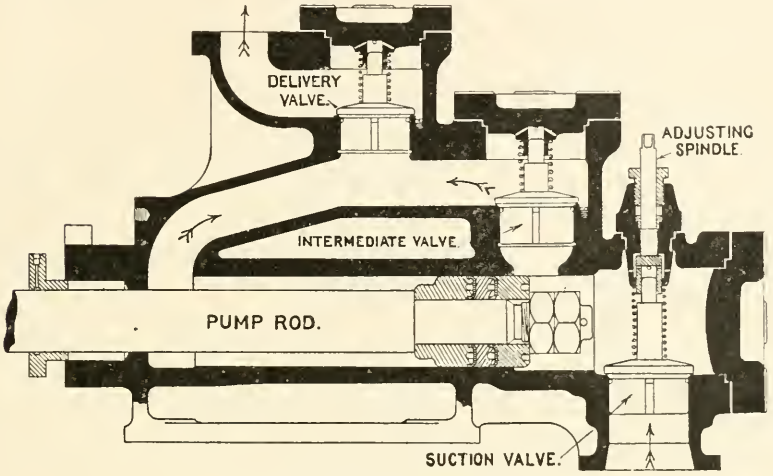


FIG. 408.—Hydraulic Power Pump.

during the inward stroke, from left to right, the suction valve closes automatically and the water is forced through the intermediate valve and fills the space around the pump rod and below the delivery valve. The pump chambers being now entirely filled with water, the plunger moves from right to left and twice as much water is drawn in through the suction valve as is delivered through the delivery valve; from

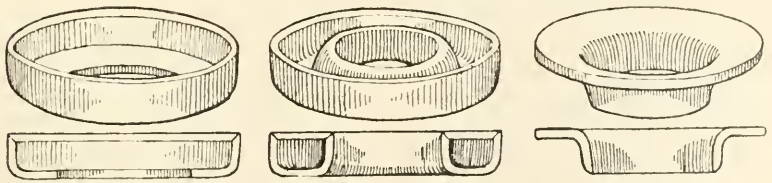


FIG. 408A.—Hydraulic Packing Leathers.

left to right the plunger forces twice as much water through the intermediate valve as is required to fill the space around the rod. There is thus an almost continuous and steady delivery of hydraulic pressure during both the forward and return strokes, although the suction is

only open during one of them. Very important matters in the design of hydraulic machinery are: the avoidance of air pockets in pump chambers and pipes; a fair run of piping without abrupt changes in direction or sectional area of the moving water; and a steady and continuous flow without change of momentum and consequent shock and loss of energy.

Hydraulic Pressure Control and Governor. — Referring to Fig. 409, the crank shaft rotates the *speed governor shaft* by means of a pair of bevel wheels B and C; the governor shaft is revolved at about two and a half times the rate of revolution of the crank shaft. When the engine speed is excessive, such as may be caused by a burst pressure main pipe, or by too many hydraulic motors at work at the same instant and thus developing an excessive power, the speed governor comes into play. The weights D tend to be thrown by centrifugal force away from the shaft, and this force, overcoming the springs E, carries the hollow spindle F to the right. F by means of a sleeve coupling acts on the lever GHJ, which is pivoted at H, and forces J to the left. J carries the rod KL with it and operates on the bent lever LMN pivoted at M, which, pulling the valve rod N downwards, tends to close the throttle-valve and reduce the steam-supply to the engine.

A nearly *constant pressure* is maintained by means of the *hydraulic governor*. Pressure from the main is admitted through a pipe P to the under side of a hydraulic ram Q, which is connected by a crosshead at the top end with two springs in compression. When the hydraulic pressure exceeds that for which the springs are set, the ram Q rises and carries with it the lever RST, pivoted at R, and thus raises the servo-motor or auxiliary valve attached to the spindle T; pressure is thus admitted below the plunger or ram W, which is forced upwards and carries with it the connecting rod WX (not shown) and lever XKJ, which pivots on J; K is thus moved to the left, and tends to close the steam throttle-valve and reduce the speed of the engine.

The movement of the ram W corresponds exactly to that of the servo-motor valve on T, and when the hydraulic pressure is too low, the ram Q lowers and carries the servo-motor with it; the pressure escapes from below the ram W, through the exhaust cavity in the servo-motor and a central hole in the ram, into an exhaust space which is in connection with an exhaust return pipe. The position of the nut N on the throttle-valve spindle is adjustable, as are also the hydraulic spring compression and length of the rod KL.

Hydraulic Gun-training Engine.—Fig. 410 shows a type of hydraulic engine used for training the barbettes or turrets in which

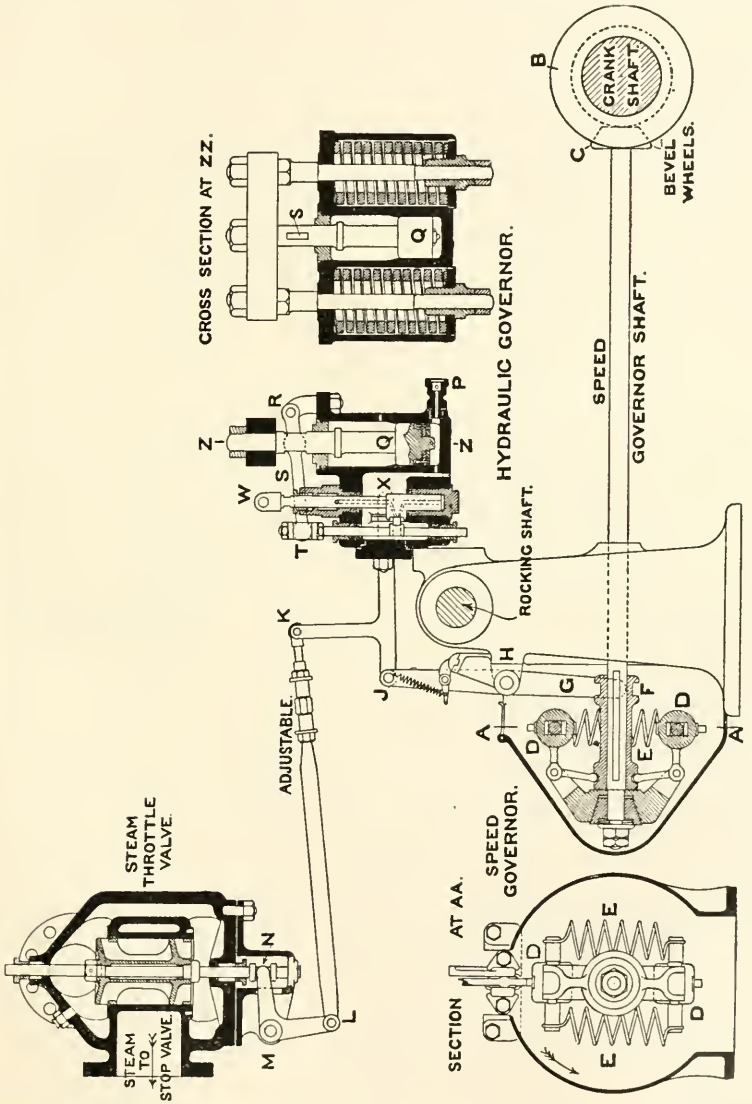


Fig. 409.—Hydraulic Governor.

heavy guns are mounted, and Fig. 411 shows a section through one of the slide valves *F* and *engine working* (differential reversing) *valve H* of the same engine.

The engine consists of three single-acting oscillating cylinders A (the middle one of which is shown in section in Fig. 410) mounted

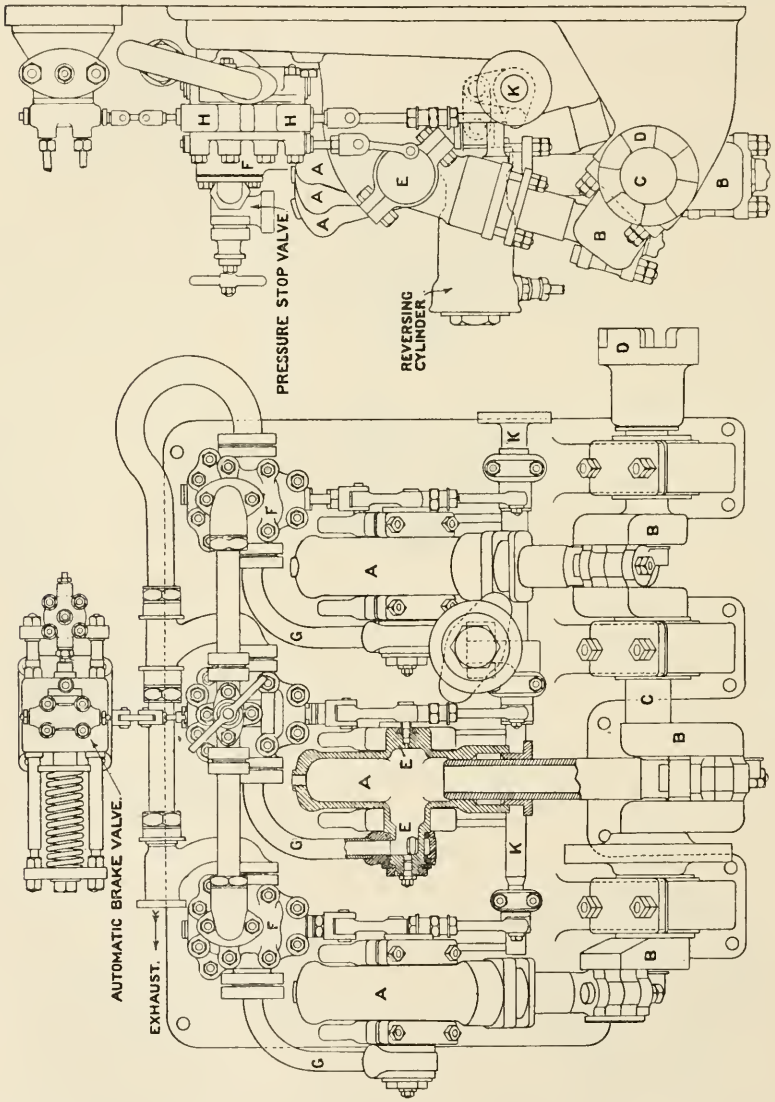


Fig. 410.—Hydraulic Gun-training Engine.

side by side on trunnions E, and acting on three cranks B set at equal angles (120°) on the same shaft C, which is connected by a coupling D and gearing (not shown) with a rack on the turret wall. In the older

ships the training engine is situated outside the turret; but in the more recent ones the engine is fitted in a turn-table below the turret, and moves with the turn-table and turret which are rigidly connected together. Usually two engines are fitted for each turret, either one of which can be coupled or uncoupled as required. Each cylinder is connected, through one of its trunnions E, a pipe G, with a slide-valve F, and working (differential) valve H; and by means of a reversing lever working a servo-motor valve in connection with the reversing cylinder, the engine can be reversed or stopped as required. The weigh shaft, or reversing shaft K, is operated by the reversing cylinder, and is coupled by levers and connecting rods with the valve H. The action is precisely similar for each cylinder and set of valves F and H; but each cylinder, being single acting, is in turn opened to

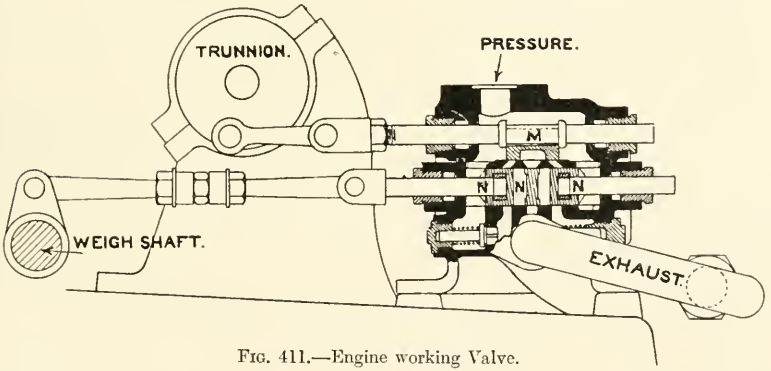


FIG. 411.—Engine working Valve.

exhaust, and the motion of the shaft is continued by the force imparted to it from the other two cylinders. Should the pressure in the system fall below a safe working limit, the brake valve comes automatically into action and, overcoming the action of the reversing cylinder, centres the valve H, and by thus blocking the exhaust, prevents any further movement of the engine. The brake valve is particularly necessary for boat and other hoisting engines when of the hydraulic type above described.

In Figs. 410 and 411 the letters F and H correspond to letters M and N respectively.

Referring to Fig. 411, the valve M is shown in the middle of its stroke, and its corresponding plunger would then be at one end of its stroke. The weigh shaft is in the position corresponding to *stop*, and consequently the valve N is in the middle of its stroke with both

pressure and exhaust ports closed. Suppose N is moved to the right, then no motion will follow, because the valve M covers both ports in the working face interposed between M and N. But the movement of the weigh shaft brings one of the other cylinders into play and sets the engine in motion, and moves M either to right or left as the case may be. Suppose M moves to the left ; then pressure, passing through the false face and the right-hand port in N into the right-hand port below N, which is in connection with the cylinder, pushes the plunger outwards. Next, suppose N is moved to the left and M moves to the left, as before, under the influence of the other cylinders ; then the exhaust cavity in the valve M connects the left-hand port below N with the exhaust, and, allowing the plunger to move back into its cylinder on the return stroke, the direction of motion of the engine is reversed.

The *Bollard engine* (hydraulic) acts on exactly the same principle as the engine above described, and the details are similar, but the cylinders are differently arranged. This arrangement corresponds to the three-cylinder Brotherhood engine generally found in torpedoes. All three plungers act on the same crank, and the cylinders are fixed at equal angles (120°) about the shaft. This type is generally used for small powers, hoisting ammunition, etc., and is more compact than that shown in Fig. 410.

The mechanical efficiency of a hydraulic engine, in combination with loss of efficiency in transmission, is low and generally less than that of electric power applied to the same purpose. Notwithstanding this low efficiency, there are many advocates of hydraulic power for working guns and hoisting ammunition in the British Navy ; but in other navies the electric system is becoming increasingly adopted, although not entirely successfully. It should be remembered, however, that electricity for working heavy guns is comparatively in its infancy, and that during a period of over forty years the hydraulic system has been steadily improved. Hydraulic power is heavier than electric, but it adapts itself more readily to variations of speed and load, and with men better trained in its use than with electricity repairs can probably be more readily made. Apart from these reasons, it is well to remember that electric power is cheaper, and of less weight, and occupies less space than hydraulic power, but hydraulic power has served us extremely well throughout the war.

CHAPTER XXXII

ELECTRIC GENERATORS AND OTHER ELECTRICAL MACHINERY

Electrical Units and Power Measurement.—It is not proposed to give an explanation of the theory of electricity, which is dealt with in the text-books on this subject, but simply to give a few definitions and units with a short explanation of their uses as applied in ordinary ships' outfits, which primarily consist of electric generating sets for lighting, ventilation, and heating. In submarines electric power is applied through secondary batteries as a motive power for propulsion, and this will be given some slight attention.

Electric generating plants are now usually specified for an output in kilowatts (K.W.) at a pressure in volts (E), which is also known as E.M.F., electromotive force, or D.P., difference of potential.

The rate of flow of the current is measured in ampères, and denoted by C, and the resistance to flow is measured in ohms, and denoted by R.

A michrom is one-millionth part of one ohm, and a megohm is one million ohms.

The watt is the lower unit of electrical power, and is equal to $C \times E$, or C^2R.

The kilowatt, or 1000 watts, is the higher unit of electrical power, and is equal to 1·34 horse-power per *minute*.

One horse-power is equal to 746 watts, or 0·746 kilowatts.

The output of a dynamo or motor in horse-power

$$= \frac{\text{ampères} \times \text{volts}}{746} = \frac{C \times E}{746} = \frac{C^2R}{746}.$$

$$\text{Ohm's law,} \qquad C = \frac{E}{R},$$

and if a dynamo gives out 400 ampères at a pressure of 225 volts, its

kilowatt is 90, and the resistance necessary to produce the desired voltage of 225 is $\frac{9}{16}$ths, or 0·5625 ohms.

The resistance determines the size of the wire, and if necessary in the case of searchlights, the amount of artificial resistance required by the lamp to produce the light, E.

Incandescent lamps will always light up regardless of the polarity, or direction of flow of the current. But searchlights require a definite polarity which can be simply tested by partially immersing two lead plates in a jam-pot containing a very weak solution of sulphuric acid and connecting the leads by wires with the terminals of the dynamo ; the *positive* plate turns brown. When charging storage batteries the correct polarity is of great importance, because if incorrect the dynamo becomes a motor, and will be considerably accelerated and run the batteries down instead of charging them. For this reason it is absolutely necessary to obtain a higher pressure, or voltage, at the dynamo terminals than in the batteries before switching on for charging, or the same thing may happen and the dynamo become depolarised.

The essentials for good and efficient running of dynamos are :

(1) Perfect cleanliness and good ventilation.

(2) Adjustment of the brushes and alteration of pitch to suit the load.

(3) A bright commutator running *true.*

(4) A well-insulated circuit.

It should be remembered that a dynamo or electric motor warms up to its work, and that in most specifications a limit of temperature is allowed above the temperature of the compartment in which it is placed for working. Large dynamos take about one hour for each 100 ampères to warm up to their normal working temperature as a maximum, but this differs somewhat according to various types of machine and the load imposed on it.

The fuel or steam consumption is generally specified at the rate per K.W.H. (kilowatt-hour), or simply per H.P.H. (horse-power-hour). In either case it refers to the H.P., or output as measured at the dynamo or motor terminals, and not to the I.H.P. or S.H.P., as the case may be, which the engine develops, and which is larger than the output of the dynamo or motor.

The proportion for reciprocating engines is generally about one-third greater, and for turbine engines about one-fifth greater than the electrical output ; the loss being due to mechanical losses in the engine

and to heating and other losses in the armature and commutator, and to friction of the brushes.

In submarines, when on the surface, electricity may be generated by utilising the prime mover, either an internal combustion engine or a steam engine, to drive a dynamo which charges secondary batteries or cells. Each cell has a capacity of 30 or more ampères according to its surface exposed to chemical action, and a pressure in the usual case of 1·8 to 2·4 volts.

The average output when discharging may be a little over 2 volts for each cell, and the average current is then about $\frac{1}{4}$ ampère for each square inch of surface. But these figures are not always applicable, as cells differ considerably in construction and working.

The dynamo is used as an electric motor for propelling the boat when submerged, and is supplied by the batteries with current at a practically constant voltage which is obtained by joining together a sufficient number of cells in *series*, that is, by joining the positive terminal of one cell to the negative terminal of the next. If the internal resistance of each cell is r, and the number of cells in the group connected in series is n, then

$$\text{the voltage} = \frac{n\mathrm{E}}{r},$$

and the current supplied by this group to the motor, whose resistance is R, is

$$\mathrm{C} = \frac{n\mathrm{E}}{r + \mathrm{R}}.$$

The shunt winding resistance of the motor may be modified to obtain a fair variation of speed when used for propelling purposes, by using a rheostat or shunt regulating resistance, and in this way the current output may be regulated and controlled; and according to the regulation the group of cells is discharged at a known rate in either a long or a short time. In other words, a certain known quantity of energy is stored in the batteries, and according to the power taken out the batteries will continue to supply energy for a long or a short time, similarly to a steam boiler for which a limited quantity of coal or other fuel energy is available.

So long as the voltage is suitable to the motor, any number of cells can be joined in series for supplying power current, but usually the cells are grouped together so as to obtain approximately the same voltage from each group, and such groups are joined in *parallel*, that is

all the positives joined in one terminal, and all the negatives in the other terminal for connection with a switchboard. From the switchboard it can be arranged for one or more groups to be connected with the motor, while the other groups are held in reserve.

Series wound motors increase speed when the magnetism of the field magnets is reduced, and are frequently fitted where constant speed is unnecessary.

Shunt wound motors give practically constant speed at all loads, but by using a rheostat, as stated above, the speed can be varied.

Compound wound motors run at absolutely constant speed at all loads ; the series wire being wound in opposition to the shunt wire. The dynamos of electric generating sets are usually compound wound, and therefore at all loads the required constant speed may be easily maintained by a well fitted and balanced governor fitted on the engine, but in some particular instances an electrically operated governor is used.

Dynamo Engine.—In Fig. 412 a two-stage expansion Brotherhood engine, used for driving an electric generator, is shown, and is of about 150 I.H.P. The H.P. slide valve is of the cylindrical type, and two valves of similar construction are used for the L.P. cylinder. All the working parts are proportioned for high-speed running, generally about 400 revolutions per minute.

A *governor* (an enlarged section of which is shown in Fig. 413) is necessary to all dynamo engines, as when working searchlights and motors the load is continually varying, and unless a fairly efficient governor is fitted, the sudden increase of the speed of the engine may produce a sufficient increase of voltage in the dynamo to fuse the armature and other winding wires. In the figure the arms *A* are attached to the crank shaft, and revolve with it. Right-angled levers are pivoted on the ends of the arms, and the ends *B* are in sliding contact with a sleeve piece attached to a small shaft projecting axially from the crank shaft. As the arms move about the pivots the small shaft is moved to and fro. The other ends of the arms *C* are fitted with weights, which are restrained from flying outwards, when the engine is in motion, by a spring, *D*, tending to draw them together.

When the speed of rotation of the engine is too great, the centrifugal force of the weights overcomes the spring, and, moving the small shaft to the left, presses on one end of a right-angled lever, pivoted at *F*. The other end of this lever is attached to the rod *G* and valve *L*. When the rod *G* is pulled downwards, it brings the valve with it,

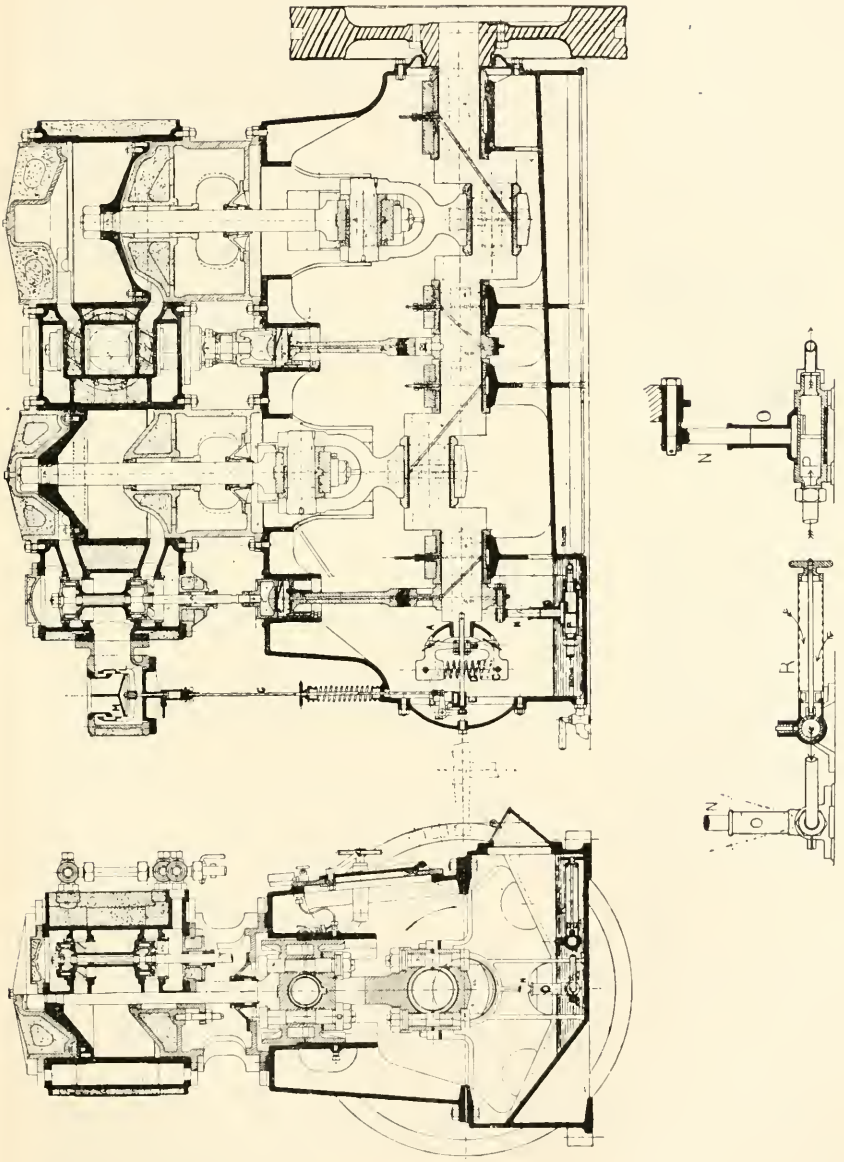


FIG. 412.—Brotherhood Two-stage Expansion Dynamo Engine.

and closes the opening on the circular fixed seating M. The valve L regulates the steam supply to the engine, and is worked entirely by the governor. An ordinary shut-off valve is fitted, above the valve box M, in the steam supply pipe. The governor valve rod is of small diameter, and being made a good fit in a long gland, prevents almost entirely all leakage.

For Brotherhood air compressors, the speed governor is of similar construction, but in addition a pressure governor, shown in Fig. 413, is also fitted. S is in open communication with the air reservoir, and when the pressure is excessive it lifts the small plunger T and admits pressure above the large plunger V, which tends to close the valve L.

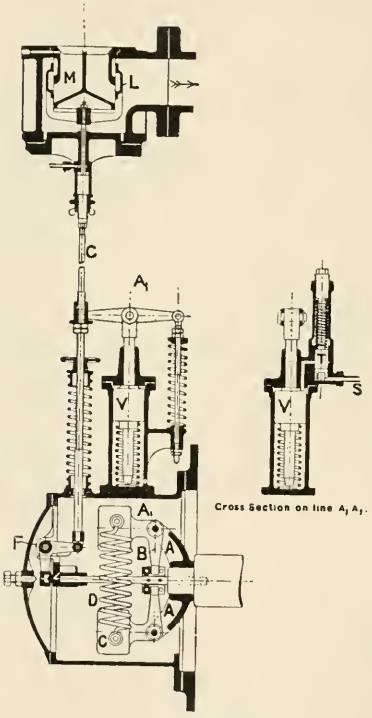


FIG. 413.—Brotherhood Combined Speed and Pressure Governor.

Forced lubrication is fitted to all the working bearings. The pump is valveless (Fig. 412), and consists of a solid plunger N, working in an oscillating pump chamber O. The suction is immersed in oil, which is strained on its way to the pump, where it enters the cylinder P, which is fixed in position. The oscillation of the chamber O is such that on the up-stroke of the plunger the oil can enter it, through a slot in the outer part of the fixed cylinder. On the down-stroke the oscillation of the pump chamber closes the suction port and opens another port in connection with a fixed discharge cylinder. From the discharge cylinder the oil is forced, under a pressure of about 40 to 50 lb. per square inch, into the various bearings.

The oil under compression enters a groove cut circumferentially about the middle of the main bearings, through small pipes fitted above the bearings, and is diffused over the whole of the bearing surface. A hole is drilled, as shown, from a main bearing to each

crank head, and the oil passes through this hole into a similarly cut groove round the crank-head bearing, and is diffused over the bearing surface. All the bearings are made with oil-containing strips, entirely enveloping the ends of each bearing, but between these strips the usual clearance spaces are left near the lips of each of the top and bottom brasses. The bearing surfaces of the eccentrics are lubricated in a similar way, and a hole through the centre of the eccentric rod conveys the oil into the top-end bearing. The crosshead brasses are lubricated through a separate pipe, directly connected with the pump delivery. The oil is injected, through holes in the back and front of the guide, into recesses, which are always covered by the guide, made in the slipper opposite the mid-length of the crosshead pin. The recesses are in direct communication with the groove cut around the crosshead brasses. Another connection serves to lubricate each guide, back and front.

The oil pump is single acting, and the supply of oil should be continuous. A relief valve is therefore fitted in the upper part of the delivery pipe, which allows the pressure to escape into a small chamber, which, acting as an accumulator, maintains an almost constant pressure in the various supply pipes led away from it to the bearings.

The oil gradually works its way through the ends of the bearings, falls into the crank pit, and, draining back to the oil-pump suction, is used over again after passing through the strainer. The strainers are of special design, and two are generally fitted, so that one can be removed and cleaned at a time. In the figure, R is the strainer, made of fine wire gauze, fitted over brass end pieces, joined together by a screwed rod, and is removable by unscrewing the rod. When the end of the strainer is lifted it automatically closes the suction opening, and the wire gauze can be removed and cleaned. When the strainer is replaced and turned down into the horizontal position, the suction is automatically opened.

The whole of the lower part of the engine is enclosed by the framing, but doors are fitted for access to all the working parts. Water dripping, or working, through the piston-rod glands is prevented from getting inside this closed chamber by shallow glands, packed with soft packing. The governor end of the chamber is closed by a removable cover for access to the governor. The flywheel end is fitted with a disc, to throw off the oil and prevent it working along the shaft; and a fixed dish-shaped flange is fitted to catch the oil as it is thrown off, and to return it by gravity to the bottom of the chamber.

Forced lubrication is usually fitted to all fast-running auxiliary

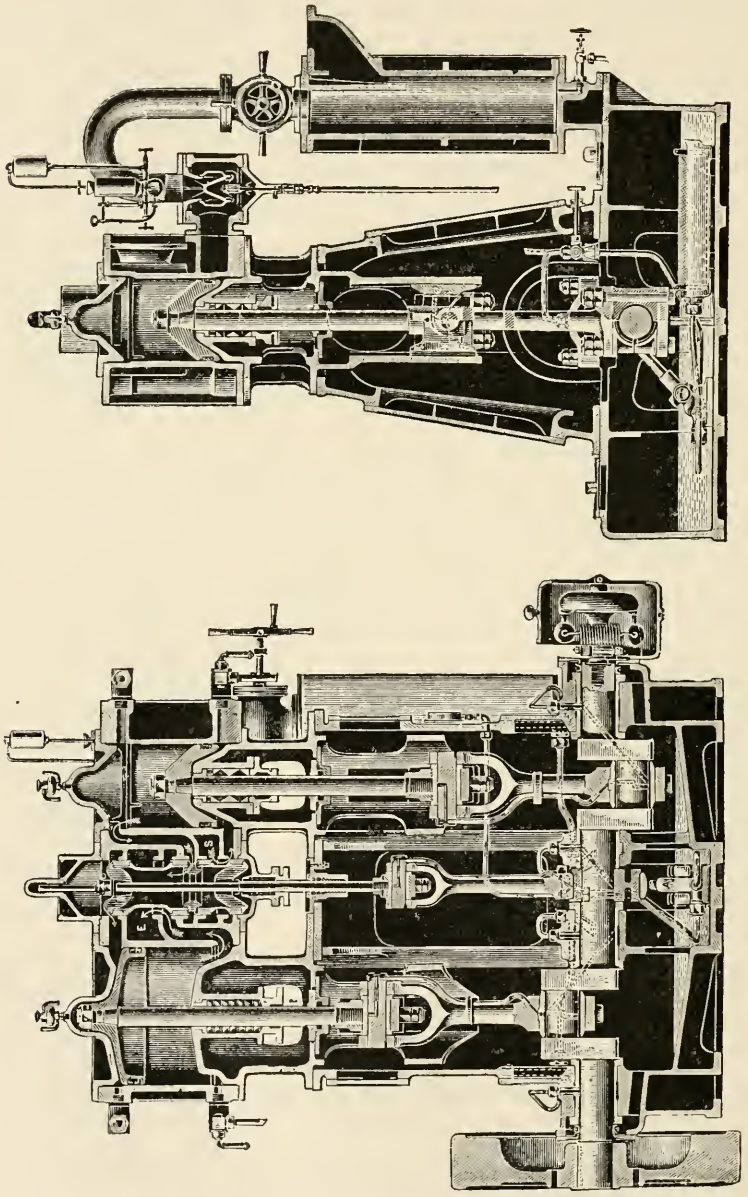


Fig. 414.—Belliss Two-stage Expansion Engine.

engines, and is now fitted in many vessels for the main propelling

engines, as a result of experiments in a destroyer. Some dynamo engines have been running for some years without appreciable wear of the bearing surfaces.

In the *Belliss and Morcom* two-cylinder *dynamo engine* only one eccentric is employed to drive the two slide valves, which are in one piece, and attached to the same rod as shown in Fig. 414. The disposal of the cranks allows this to be done without any complications. The cranks are always fitted opposite to each other in the two-cylinder engines used for dynamos. This produces less vibration and more uniform running, as the reciprocating weights are more nearly balanced. Messrs. Belliss and Morcom are, I believe, the pioneers of forced lubrication ; but the system differs a little from that above described, although the same in general principle.

Some interesting tests of a three-stage expansion dynamo engine (Fig. 415) were made by them on the effect of superheat, and they found there was a gain of—

10	per cent	in	economy	by superheating	50° Fahr.
$17\frac{1}{2}$	,,	,,	,,	,,	100° ,,
23	,,	,,	,,	,,	150° ,,
$27\frac{1}{2}$	,,	,,	,,	,,	200° ,,

with a corresponding total reduction of the weight of steam used per kilowatt-hour of $6\frac{1}{2}$ lb. (from $23\frac{1}{2}$ to 17 lb.).

Engine Power required for Driving a Dynamo.—The electrical horse-power of a dynamo is equal to the current (in ampères) multiplied by the difference of potential (in volts) and divided by the number 746. The over-all efficiency may be taken as about 66 per cent, and therefore

$$\text{I.H.P. of engine} = \text{ampères} \times \text{volts} \div 746 \times 66 \text{ per cent}$$
$$= \text{ampères} \times \text{volts} \div 500, \text{ about.}$$

EXAMPLE.—Suppose a current of 1000 ampères and a voltage of 120 is required; then—

$$\text{I.H.P. of engine} = 1000 \times 120 \div 500 = 240.$$

The approximate mechanical efficiency of any reciprocating dynamo engine is easily found, because the electrical horse-power is always obtainable from the instruments supplied, and the I.H.P. can be obtained by taking diagrams.

The construction of the auxiliary condenser is described in Chapter XXIV..

Parsons Turbo-Electric Generating Engine. — In several

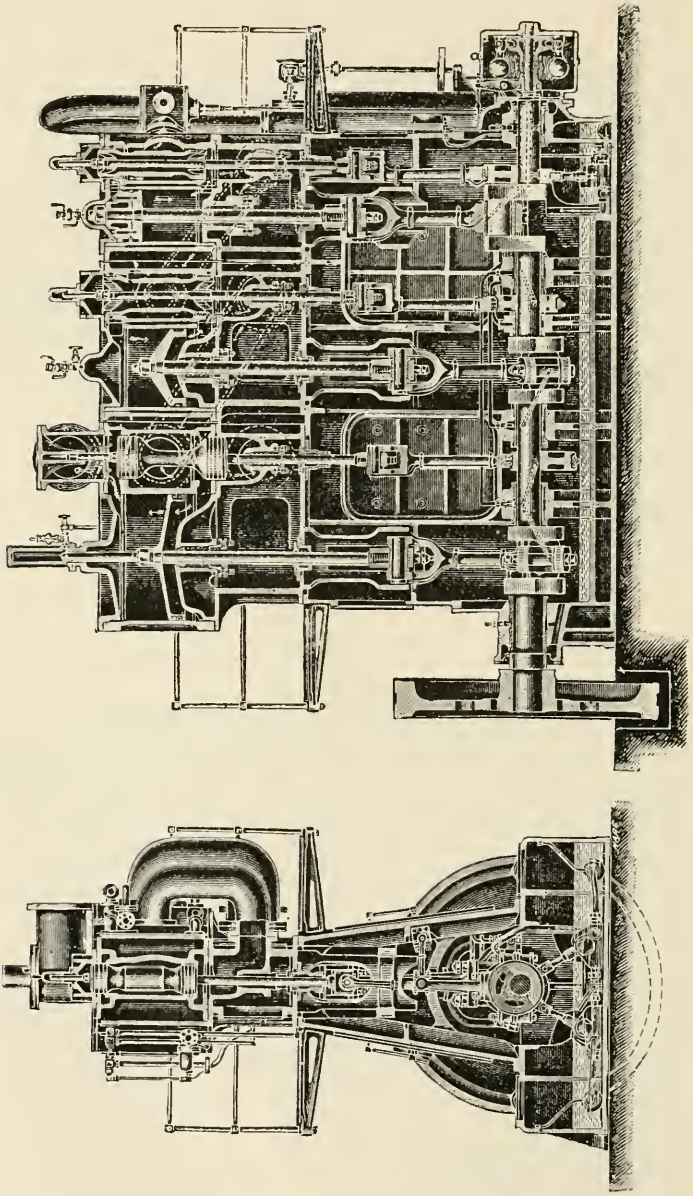


FIG. 415.—Belliss Three-stage Expansion Engine.

recently constructed vessels turbo-electric generating engines are

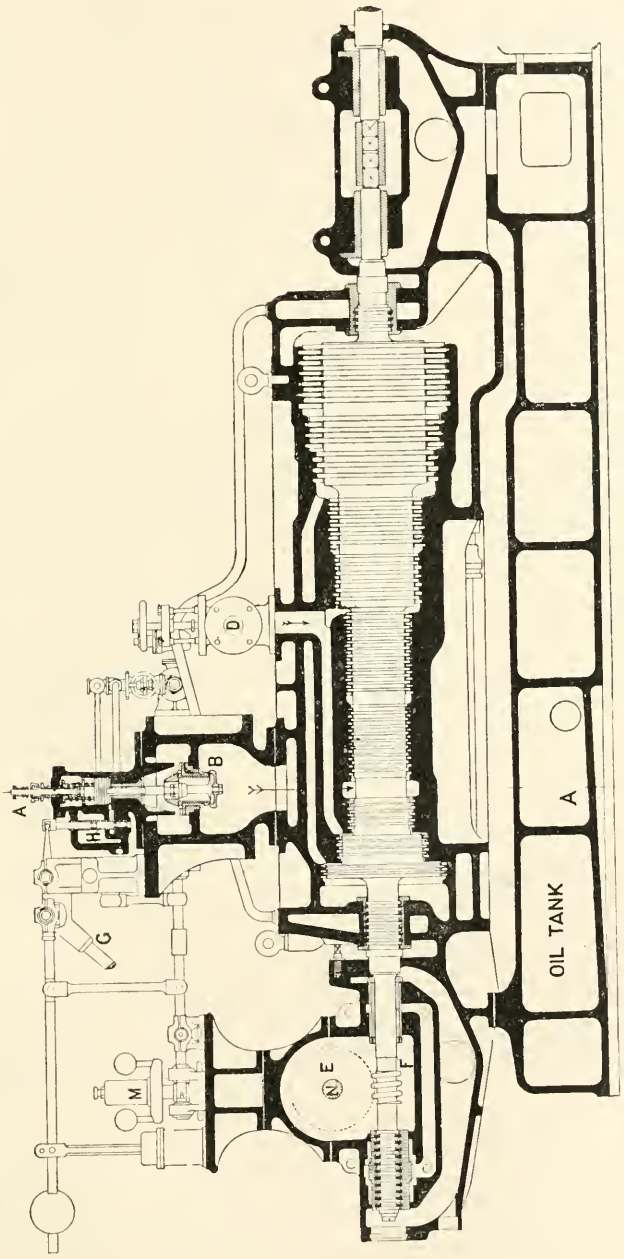


FIG. 416.—Turbo-Electric Generating Engine.

fitted. The Parsons reaction turbine shown in Fig. 416 is one of this type, and its action will be better understood after studying Part VI. In the meantime, the more important accessories will be described. The speed governor *M* (and gearing illustrated in Figs. 416 and 417) is of the usual centrifugal type, and is rotated at just below one-tenth the rate of the engine shaft by means of a worm wheel *E* and screw *F*, and by bevel gearing (not shown) driven by the shaft *N*. This shaft, by means of a crank, drives a forced lubrication oil pump, and also a belt pulley for a tachometer, as well as a second pair of bevel wheels and a second governor. One governor controls the total supply of steam through the valve *C*, while the other admits this supply intermittently (in gusts) at full pressure through the valve *B*. Both the valves, *B* and *C*, are of the balanced or double-beat type. Above the valve *B* is a cylinder and piston; the piston is pressed downwards by a strong spring, while the steam pressure below tends to force it upwards and thus close the valve. Steam pressure gains access to the space below the piston through a clearance space around the valve-spindle when the valve is opened, but when the valve is closed this clearance passage is also closed. Movement of the rod *G* by the governor-driven gearing produces corresponding movements in the servo-motor (or auxiliary) valve *H*, which, opening or closing the exhaust from below the piston, produces corresponding movements in the valve *B*. Thus greater movement of the governor produces a longer opening

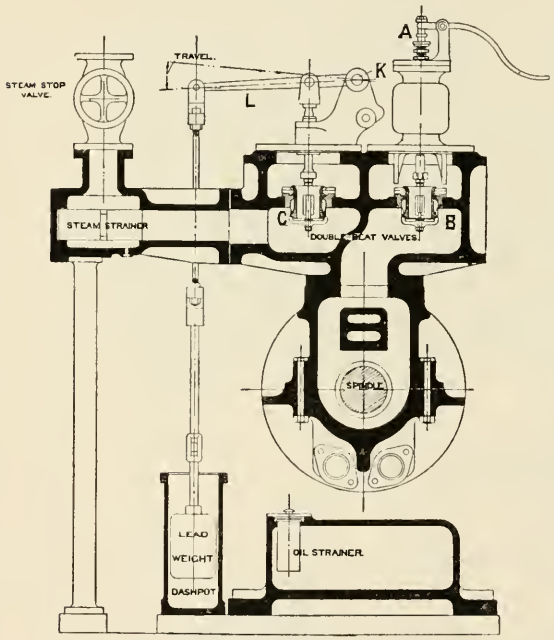


Fig. 417.—Turbine Speed Governor and Regulator.

of the valve B. The valve B can be held open, if required, by means of the hand lever shown, and excessive working loads can

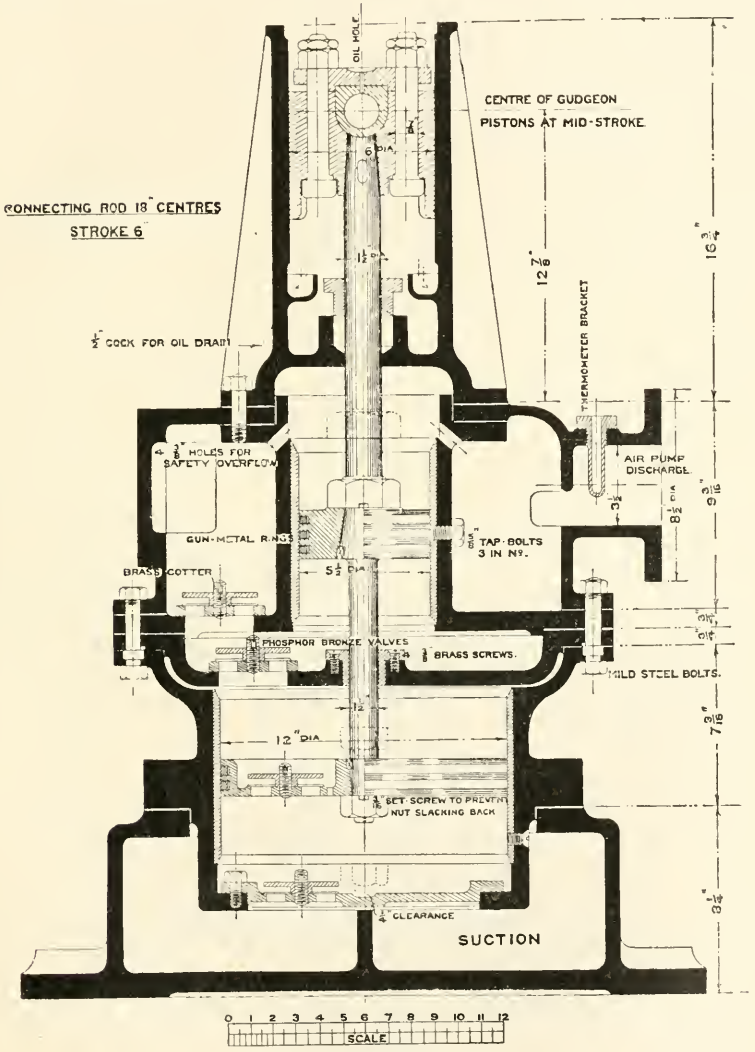


FIG. 418.—Air Pump for Turbo-Electric Generating Engine.

be carried by the engine by admitting steam directly through D to a higher stage of expansion.

The valve C is operated by the governor sliding connection or muff

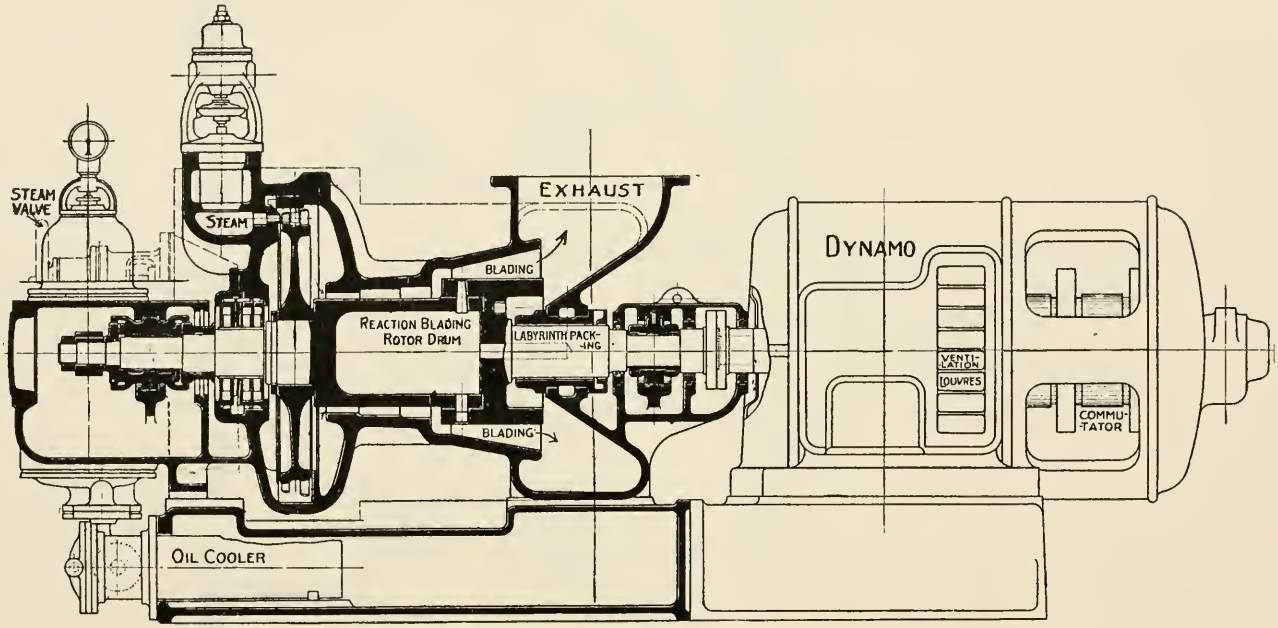


FIG. 419.—Brown-Boveri Turbo-Electric Generator (Elevation).

and lever acting on the shaft K and lever L, and the oscillation is steadied by a dash pot and dead weight shown in Fig. 417.

As explained earlier, a high vacuum is necessary to the economy of turbines, and for this reason a turbo-generator is generally fitted with a separate condenser, circulating pump, and air pump. Fig. 418 shows a section through the air pump fitted in connection with the turbine shown in Fig. 416. There are two plungers of different diameters fitted on the same rod. The upper plunger is solid, and the lower one is of the bucket type. The upper plunger ensures the proper closing of the lower pump delivery valves, which, being water-pressure sealed on the downward stroke, do not leak, and thus assists in maintaining a high vacuum by the bucket pump.

This sketch is fully dimensioned and serves as an example of a fairly good working drawing, which is worth studying and is shown here principally for that purpose.

B.B.C. Turbo-Electric Generator.—Fig. 419 shows a sectional elevation, and Fig. 420 an end view in part section of a Brown-Boveri Impulse-reaction turbine of a type rated for 100 to 300 K.W. output at the dynamo when running at 4500 to 3600 revolutions per minute according to the size and design. The impulse wheel shortens the length compared with entire reaction blading, reduces to a minimum the risk of blade stripping, and the critical speed of the shaft is well above the normal speed.

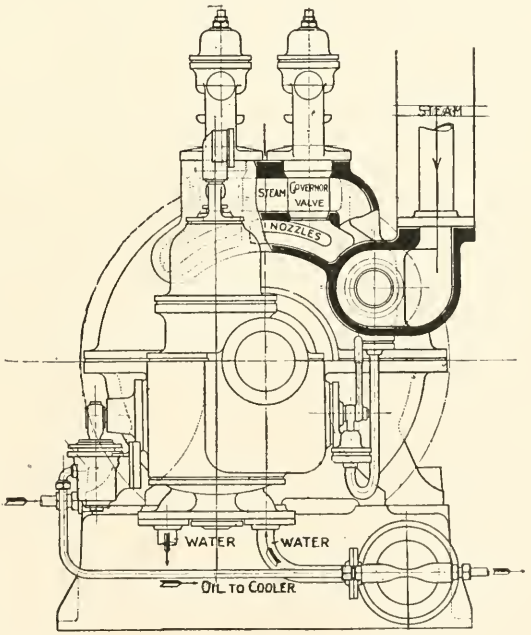


FIG. 420.—End view of Fig. 419.

Regulation of the steam supply is effected by oil distribution under

pressure, and the speed governor is also arranged to actuate a safety device which closes the main steam valve when the normal speed is exceeded by a certain amount. The larger of these turbines is provided with a nozzle regulator which automatically opens or closes the nozzles, and is controlled by the oil pressure system.

To make the generator independent of other auxiliary machinery the set is provided with a special oil pump and a separate water pump for cooling the oil. The water pump is of the reciprocating type and is worm driven from an extension of the governor spindle. The oil reservoir is built into the bed plate of the set. The bearings are lubricated by oil under pressure, and the outer turbine shaft bearing also acts as a thrust bearing. Carbon packing is used for the high pressure end, and labyrinth packing for the exhaust end of the shaft.

The steam consumption of the larger sets with reaction blading is more economical than reciprocating engines working under the same conditions of steam and exhaust pressures, and the saving in weight and space is very considerable. For instance, comparing a small 80 kw. set running at 4500 revolutions per minute with a reciprocating engine set running at 500 revolutions per minute, the weights are respectively $1\frac{1}{2}$ and 9 to 10 tons, or as 1 to 6. There is a large field for the adoption of small turbines for auxiliary work of every description, and as the dynamo is the most constant in use it is of the greatest importance that it should be at least as economical as the older type of reciprocating engine. Weight and space are in this instance of secondary consideration, because any difference may be easily counterbalanced by greater steam and fuel consumption, but if the steam consumption is equal to or less than that of a reciprocating engine then the turbine is undoubtedly the best to adopt, because its upkeep is less and its efficiency does not fall off with age and wear.

CHAPTER XXXIII

REFRIGERATING MACHINERY

THERE are three principal systems of refrigeration, or the mechanical production of cold, in general use :

1. The ammonia, or NH_3, compression system.
2. The carbonic anhydride, or CO_2, compression system.
3. The air compression system.

The objects of a refrigerator are to reduce the temperature in a cold chamber, and to maintain the lower temperature for long periods.

For the purpose of preservation and storage of provisions in ships' refrigerating chambers, the following temperatures are recommended for various classes of provisions by Messrs. J. & E. Hall, Dartford :

Fruits	36 to 40° F. or	2 to 5° C.
Vegetables	36 „ 40° F. „	2 „ 5° C.
Chilled meat	29 „ 30° F. „	−2 „ −1° C.
Frozen meat	16 „ 24° F. „	−9 „ −4° C.
Frozen poultry	18 „ 24° F. „	−8 „ −4° C.
Frozen fish	18 „ 24° F. „	−8 „ −4° C.
Butter	18 „ 24° F. „	−8 „ −4° C.
Cheese	36 „ 40° F. „	2 „ 5° C.
Eggs	32 „ 35° F. „	0 „ 2° C.
Tinned goods, meats, and fruit	36 „ 40° F. „	2 „ 5° C.
Beer, wines, etc.	40 „ 45° F. „	5 „ 7° C.
Milk may be frozen in closed cans	18 „ 24° F. „	−8 „ −4° C.

Small installations usually have two insulated chambers, as shown in Fig. 421 ; the meat room used for all frozen produce such as frozen meat, poultry, fish, or butter, and the vegetable room used for vegetables and fruits.

In the larger installations, separate chambers are usually set apart for the various requirements.

Ice-making is frequently fitted in connection with the meat chamber, but preferably a separate chamber should be set apart for this purpose.

For cooling of magazines, a temperature exceeding 70° F should not be exceeded. *All* classes of explosive deteriorate at a steadily increasing rate at temperatures above about 80 or 90° F., and as all the

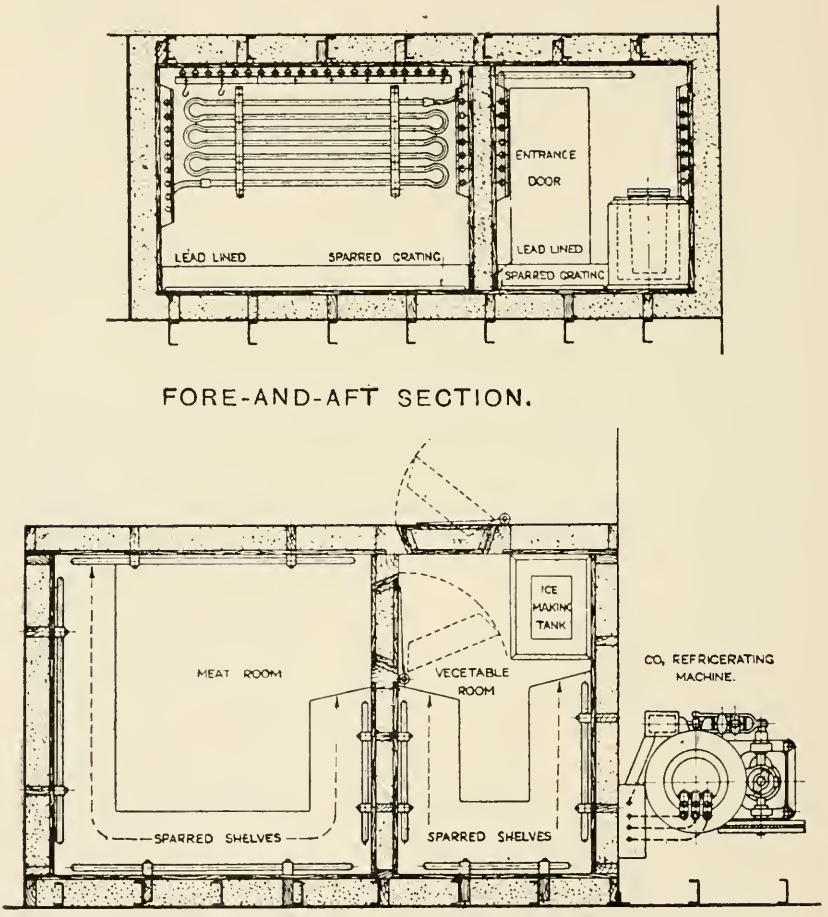


Fig. 421.—Typical Arrangement of Small Installation.

requirements for combustion are contained within the explosive itself, a process of spontaneous combustion ensues with increasing temperatures until possibly a temperature corresponding to the ignition temperature of the particular explosive is reached, when actual explosion

takes place. By maintaining a low temperature the process of combustion is retarded and almost complete safety is assured for long periods of storage. The rate of the slow combustion can be calculated from experiments, and the safe period fairly well estimated when a curve of the temperature of treatment is known.

Principle of Refrigeration.—If gas is generated from a liquid in a closed vessel there is a change of state ; and the pressure continues to rise as more gas is generated, because there is only about the same volumetric space to fill. As the pressure rises, the temperature at which the liquid boils continues to rise also, at a rate accurately determined by experiment. If we take a certain quantity of this gas, at a certain pressure, and still further compress it without adding any more heat, some of the gas returns to the liquid state, because it is then compressed above the pressure corresponding to its boiling temperature under natural conditions ; if the pressure be sufficiently increased and heat is also taken from it, the whole of the gas liquefies. This property is common to all gases, and is true of both ammonia and carbonic anhydride (carbonic acid gas).

In refrigerating machines, the less the rise in temperature during compression of the gaseous medium the greater is the efficiency, because the engine or motor has less work to do in compressing the gas. The coefficient of performance

$$= \frac{\text{Heat extracted}}{\text{Heat equivalent of work expended}},$$

and if t_0 and T_0 represent the absolute temperatures at which the machine takes in and rejects all its heat, then the coefficient of performance

$$= \frac{t_0}{T_0 - t_0}.$$

For all machines the ordinary working limits of temperature are about -4 to $+86$ Fahrenheit. The ordinary limits of absolute pressure corresponding to these temperature limits are for—

Carbon dioxide . .	288 and 1038 lb. per sq. in.
Ammonia gas . .	27 and 170 lb. per sq. in.
Sulphur dioxide . .	9 and 66 lb. per sq. in.

Some chemical and physical properties of the above three gases are shown in the following table :

	CO_2.	NH_3.	SO_2.
Critical temperature, Fahr. . . .	88·43	266·0	312·8
Critical pressure, lb. per sq. in. . .	1071·	1624·0	1159·6
Mean specific volume of liquid, cu. ft. .	..	0·0256	0·0112
Specific heat of liquid . . .	0·98	1·02	0·40
Kp (see page 273)	0·217	0·508	0·154
Kv ,, ,, ...	0·172	0·393	0·123
γ ,, ,, ...	1·26	1·29	1·25

Thermodynamically the sulphur dioxide machine is the most efficient ; the ammonia gas machine is nearly as efficient as the sulphur machine, but the carbon dioxide machine is more than 10 per cent less efficient than either of the others. Roughly, the sulphur dioxide machine is the largest, and the carbon dioxide the smallest, but the high working pressure of the latter makes the actual weights about equal. Practical objection to SO_2 machines is the inward leakage of air making for low efficiency in ice-making, but where the lower limit of pressure is atmospheric or nearly so (as in coolers for dairy work) sulphur dioxide is a most suitable refrigerant. The Board of Trade do not allow ammonia machines to be fitted in a steamer's engine-room because the gas is extremely noxious and a man cannot possibly live in the fumes. This machine is the most efficient ice-maker and, if a suitable position can be found, it is sometimes fitted for ice-making, magazine-cooling, and general refrigerating purposes. Carbon dioxide is not poisonous, but it does not support life, and being heavier than air it sinks and tends to flood any compartment into which it finds its way. An exhaust fan drawing its supply from near the floor is thus necessary, and is generally fitted, in any comparatively confined space, as part of the refrigerating plant. The great working pressures used with CO_2 necessitate very careful manufacture of the compressor and a special gland packing and jointing material. Any packing which allows the lubricating oil or its fumes to mingle with the gas causes fouling in the evaporator coils, and prevents efficient working, and is wasteful.

Rated Capacity of Refrigerators.—The Refrigeration Research Committee of the Institution of Mechanical Engineers, of which Sir Alfred Ewing, late Director of Naval Education, was Chairman, recommended—

(1) That the refrigeration produced by a refrigerating machine be expressed in *calories per second*. This unit is equivalent to 342,860 b.t.u. per day of 24 hours.

(2) That the refrigeration may be stated for particular conditions or for standard conditions.

(3) That the standard conditions be defined as follows: The temperature limits to be steady; the temperature of the cooling water to range from 15° C. (59° F.) at inlet to 20° C. (68° F.) at outlet, and the temperature of the brine to range from 0° C. (32° F.) to −5° C. (23° F.).

(4) That the refrigeration produced under standard conditions be called the *rated capacity* of the machine.

Thus, a machine may be classed as having a rated capacity of *one unit* if it produces a refrigeration of one calorie per second (say 342,860 b.t.u. per day) in steady working under the standard conditions specified above. Similarly a *two-unit* machine is one which, under the same conditions, produces a refrigeration of *two calories per second*, and so on.

This mode of rating may be taken as applying to absorption machines as well as to vapour compression machines.

In cases of cooling by direct expansion, without the use of brine, the same method of rating will apply, except that the conditions have to be modified by specifying for the lower limit of temperature that of the working substance itself.

It is considered that a temperature of −10° C. (14° F.), as a lower limit for the working substance, will harmonise with the conditions which have been laid down for machines using brine, and this figure will also be in accordance with the standards of temperature adopted by the Association Française du Froid.

The Committee accordingly recommend—

(5) That, in the rating of a direct expansion refrigerating machine, the temperature of the vapour in the evaporator be taken as −10° C. and that the conditions as regards cooling water be the same as those laid down for machines using brine.

The unit adopted in the United States is 288,000 b.t.u. per day.

To those interested in the subject of the theory of refrigeration, the Report of the above Committee can be recommended as being the most concise and simple explanation available. (See *Journal of Proceedings, Institution of Mechanical Engineers* for November 1914, from which the following tables are also taken.)

Tables showing Properties of Substances used in Refrigeration

Table I. *Carbonic Acid*

Pressure Absolute.	Temperature.		Volume of Saturated Vapour.
Lb. per sq. in.	° C.	° F.	Cubic ft. per lb.
200	−32·0	−25·5	0·463
225	−28·3	−18·8	0·409
250	−24·8	−12·7	0·365
275	−21·7	− 7·0	0·330
300	−18·7	− 1·7	0·301
325	−16·0	+ 3·2	0·276
350	−13·4	+ 7·9	0·255
375	−10·9	+12·3	0·236
400	− 8·6	+16·5	0·219
450	− 4·3	+24·3	0·191
500	− 0·3	+31·4	0·169
550	+ 3·4	+38·1	0·150
600	+ 6·8	+44·2	0·134
650	+10·0	+50·0	0·120
700	+13·0	+55·4	0·108
800	+18·6	+65·5	0·0878
900	+23·6	+74·6	0·0720
1000	+28·3	+82·9	0·0565
1050	+30·5	+86·8	0·0460
1070	+31·3	+88·4	0·0352

Table II. *Ammonia*

Pressure Absolute.	Temperature.		Volume of Saturated Vapour.
Lb. per sq. in.	° C.	° F.	Cubic ft. per lb.
6	−49·2	−56·6	41·6
8	−44·2	−47·6	31·6
10	−40·2	−40·4	25·75
12	−36·8	−34·3	21·75
14	−33·8	−28·9	18·79
16	−31·2	−24·1	16·56
18	−28·8	−19·8	14·82
20	−26·6	−15·9	13·45
25	−21·8	− 7·2	10·88
30	−17·7	+ 0·1	9·17
35	−14·2	+ 6·5	7·93
40	−11·0	12·2	6·99
45	− 8·1	17·4	6·25
50	− 5·5	22·1	5·66
60	− 0·83	30·5	4·77
70	+ 3·3	37·9	4·12
80	+ 6·9	44·5	3·63
90	10·3	50·5	3·25
100	13·3	56·0	2·936

TABLE II.—(*Cont.*) *Ammonia*

Pressure Absolute.	Temperature.		Volume of Saturated Vapour.
Lb. per sq. in.	° C.	° F.	Cubic ft. per lb.
120	+18·8	+65·8	2·466
140	23·6	74·5	2·124
160	27·9	82·3	1·868
180	31·9	89·4	1·666
200	35·5	95·9	1·504
225	39·6	103·2	1·340
250	43·4	110·1	1·208
275	46·9	116·4	1·098
300	50·2	122·4	1·007
350	56·2	133·2	0·863
400	61·6	142·9	0·752
450	66·6	151·9	0·665
500	71·1	160·0	0·597
550	75·3	167·6	0·539
600	79·3	174·7	0·491
650	83·0	181·4	0·449
700	86·5	187·7	0·414

TABLE III. *Sulphurous Acid*

Pressure Absolute.	Temperature.		Volume of Saturated Vapour.
Lb. per sq. in.	° C.	° F.	Cubic ft. per lb.
5	−32·0	−25·6	14·1
6	−28·6	−19·5	11·9
7	−25·8	−14·4	10·3
8	−23·2	− 9·7	9·04
9	−20·8	− 5·4	8·10
10	−18·6	− 1·5	7·35
12	−14·7	+ 5·6	6·17
14	−11·2	+11·8	5·34
16	− 8·2	+17·3	4·70
18	− 5·4	+22·3	4·19
20	− 2·8	+26·9	3·80
25	+ 2·7	+36·9	3·08
30	+ 7·4	45·4	2·59
35	11·6	52·8	2·24
40	15·3	59·5	1·97
45	18·6	65·5	1·76
50	21·6	71·0	1·59
60	27·1	80·8	1·33
70	31·9	89·3	1·15
80	36·1	97·0	1·01
90	40·0	103·9	0·90
100	43·5	110·3	0·82

TABLE IV. *Specific Heat of Calcium Chloride Brine*

Temperature.	Density [1] 1·19. Twaddell [1] 38.	Density [1] 1·20. Twaddell [1] 40.	Density [1] 1·21. Twaddell [1] 42.
° C.			
−25	0·697	0·685	0·674
−20	0·702	0·690	0·679
−15	0·707	0·695	0·684
−10	0·712	0·700	0·689
− 5	0·717	0·705	0·694
0	0·722	0·710	0·699
+ 5	0·727	0·715	0·704
+10	0·732	0·720	0·709
+15	0·737	0·725	0·714
+20	0·742	0·730	0·719

[1] The density is measured at 60° F. (15·5° C.) (Nat. Phy. Lab. Report, 1909).

TABLE V. *Calcium Chloride Brine*

Variation of Specific Gravity with Temperature

Sample A.		Sample B.		Sample C.	
Temp. ° F.	S. Gravity.	Temp. ° F.	S. Gravity.	Temp. ° F.	S. Gravity.
60·0	1·2155	60·0	1·199	60·0	1·193
55·0	1·2165	55·0	1·200	55·0	1·194
50·5	1·2175	50·0	1·201	50·0	1·195
45·0	1·2185	45·0	1·202	45·0	1·196
40·0	1·2195	40·0	1·203	40·8	1·197
35·0	1·2205	35·0	1·204	35·0	1·198
30·1	1·2215	30·5	1·205	30·0	1·199
25·1	1·2230	25·5	1·206	25·1	1·200
20·0	1·2240	20·2	1·207	20·0	1·200
15·0	1·2250	15·1	1·208	15·1	1·201

The figures given are those actually obtained during the experiments by Mr. G. W. Daniels, B.Eng., no smoothing out of the results being attempted. These preliminary tests show that—

(1) Density decreases with increasing temperature between the temperatures used ;—
(2) The variation in density is greater the denser the brine ;
(3) For the densities tested, the variation is negligible for the temperature ranges used in practice.

It may be noted that the variation is greater than that of pure water, as shown below.

Temperature ° F.	Specific Gravity of Pure Water.	Specific Gravity of Brine.	Increase in Specific Gravity.	
			Water.	Brine.
			Per cent.	Per cent.
60·0	1·0000	1·2155	0·04	0·164
50·0	1·0004	1·2175		

Ammonia System.—Ammonia under normal atmospheric pressure and temperature exists as a gas, but if compressed to about 105 lb. per sq. inch (or about 7 atmospheres), it becomes a liquid. In other words, the boiling point under this pressure is about 60° F., while under a pressure of 1 atmosphere it is about −38° Fahr. If the liquid can be reduced to a temperature of about −110° Fahr., it freezes and becomes solid.

The general system of ammonia refrigeration is shown in Fig. 422, but the sketch does not represent any particular plant. The gas is first compressed to about a pressure of 100 to 170 lb. per square inch, according to the temperature of the surroundings and of the gas-supply.

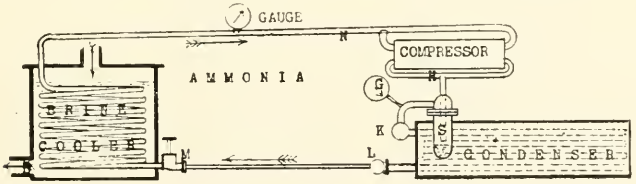
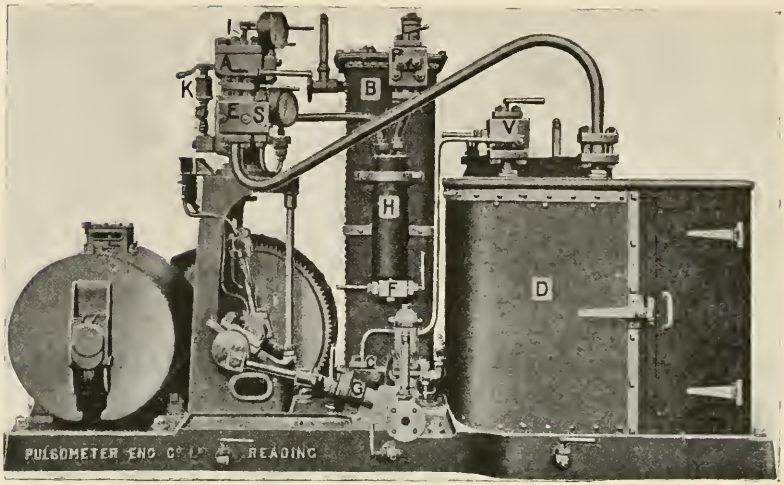


FIG. 422.—Ammonia Compression Refrigerator, Cycle of Operations.

The compressed gas is then conveyed through an oil separator S to a cooling condenser, where, the compression having already raised the boiling-point, the reduction in temperature converts the ammonia gas to a liquid. The liquid is allowed to accumulate inside the condensing coil until a slight head of liquid pressure is produced. Under this small head the liquid is allowed to expand through a small pipe from the header L to the valve M, which admits it into a coil immersed in brine. The increase in volume on entry to the brine cooling coils allows the liquid to expand and gasify at a temperature of about 10° Fahr., while the heat required for the conversion of the liquid into a gas is drawn from the surrounding coils and brine. The quantity of heat thus extracted from the brine is nearly equal to the quantity given out by the compressed gas to the condensing water during liquefaction,



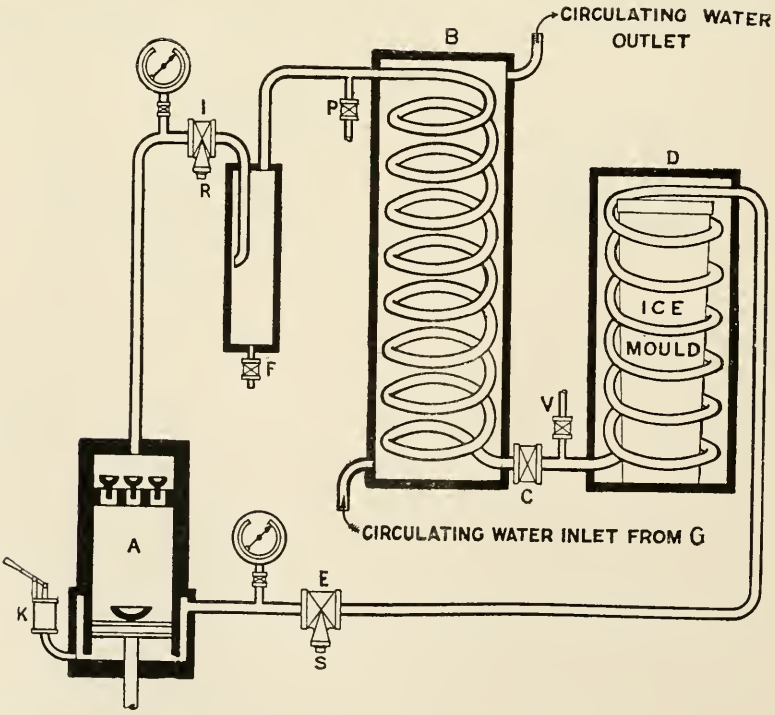


Fig. 423.—Ammonia Compression Ice-making Machine.

and the difference is equal to the work done by the steam engine in the same time. The gas returns to the compressor through the pipe *N*, and, being again compressed, commences a fresh cycle of operations.

The cooled brine is circulated by a pump through a series of pipes, arranged above and around the refrigerating chamber, and is then returned to the brine cooler for another cycle of operations. Air is sometimes used instead of brine, and is then circulated through a circuit of air trunks by a fan and returned to the cooling chamber. It is necessary to use brine, instead of water, to resist freezing; and salt is added to pure water until one gallon of brine weighs about 13 lb. (water equals 10 lb.).

Fig. 423 shows an elevation and diagram of an electrically-driven anhydrous ammonia (NH_3) compression ice-making machine made by the Pulsometer Engineering Company. Many of these machines are also steam-driven, but electrical driving is frequently more convenient. The following particulars refer to the figure:

A, Compressor.
B, Condensing tank and coil.
C, Regulating and expansion valve.
D, Refrigerating tank, coil, and ice-mould.
E, Suction stop valve (non-return).
F, Blow-off valve on *H*.
G, Circulating water pump.
H, Receiver or oil separator.
I, Discharge stop valve.
K, Oil pump.
P, Foul gas valve.
R, *S*, Drain and clearing plugs.
V, Charging valve.

The *suction* pressure, shown by the gauge near *E*, should be such that the boiling-point should be 10° F. *below* the temperature of the *brine* contained in *D*.

The discharge pressure, shown by the gauge near *I*, should be such that the boiling-point should be 10° F. *above* the temperature of the *sea* (circulating) inlet *water* at *G*.

No copper, or copper alloy, can be used in contact with ammonia.

Carbonic Anhydride System.—Carbonic anhydride, CO_2, under normal atmospheric pressure and temperature, exists as a gas, but if compressed to about 34 atmospheres (or about 500 lb. per square inch), it becomes liquid at a reduced temperature of about 30° Fahr. In other words, the boiling-point of carbonic anhydride, under a pressure of 34 atmospheres, is about 30° Fahr., while under a pressure of 1 atmosphere it is about −120° Fahr.

The general system of refrigeration is shown in Fig. 424, which does not, however, represent any particular plant. The gas is first compressed to a pressure of about 55 atmospheres (825 lb. per square inch), according to the temperature of the surroundings and of the gas-supply. The compressed gas passes from the compresser *C*, through an oil

separator S and a pipe H into the condenser coil, where it liquefies. The liquid is allowed to expand through the valve M, and, expanding

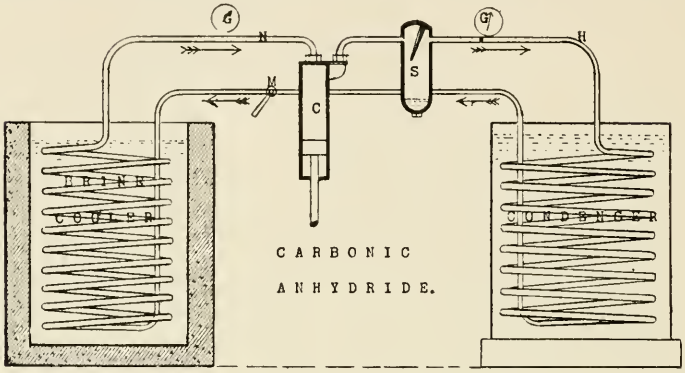


Fig. 424.—CO_2 Refrigerator.

into the brine-cooling coils, is converted into a gas while extracting from the brine the necessary heat for its conversion. The gas returns

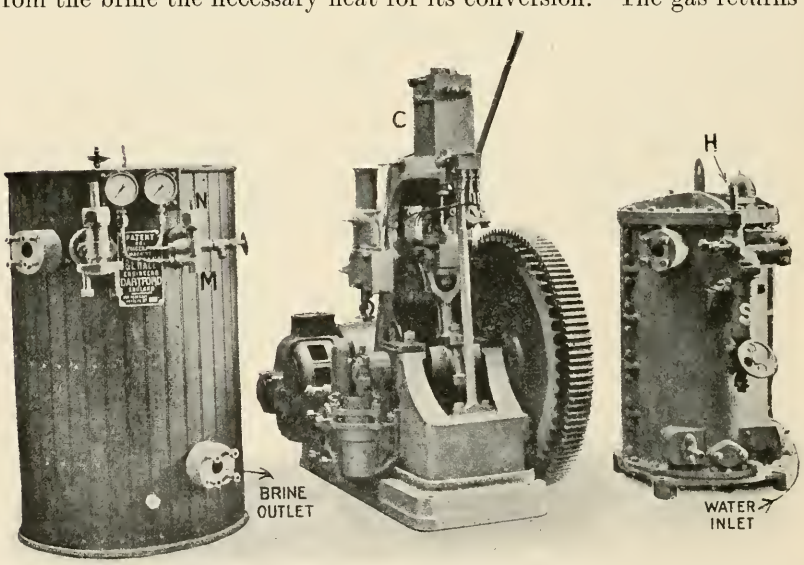


Fig. 425.—Hall CO_2 Naval Type Refrigerator.

to the compressor at a pressure of about 25 atmospheres (375 lb. per square inch). If the available temperature of the condensing water

is high, the pressure must be correspondingly high throughout the system; this is also common to all systems.

The pressures used with the carbonic anhydride refrigerator are necessarily very great, and some difficulty is generally experienced in keeping the packing glands and joints tight enough to prevent leakage.

To overcome this, Messrs. J. & E. Hall have introduced the form

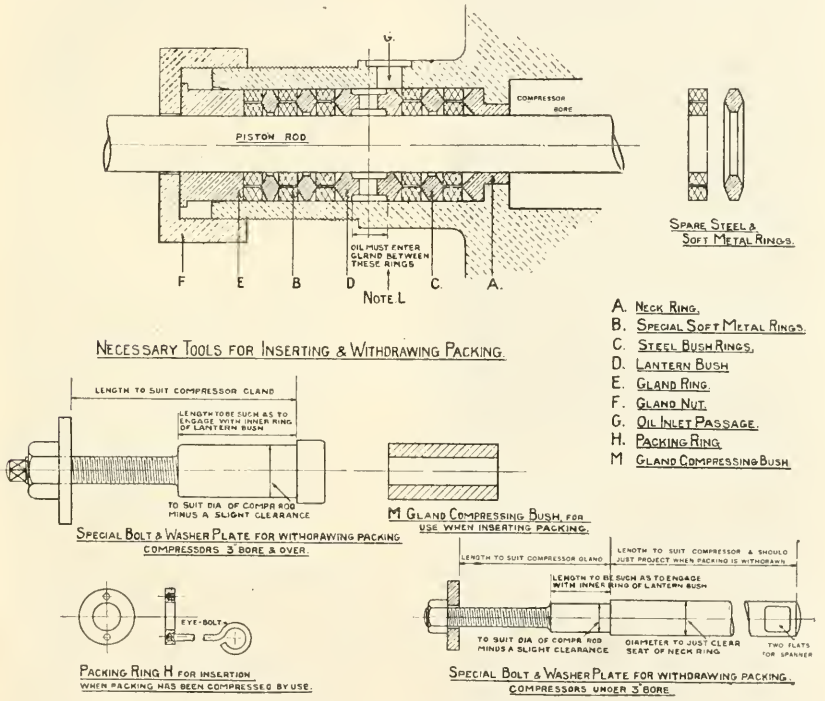

COMPRESSOR BORE
PISTON ROD
OIL MUST ENTER GLAND BETWEEN THESE RINGS
F E B D NOTE L C A
SPARE STEEL & SOFT METAL RINGS.
A. NECK RING.
B. SPECIAL SOFT METAL RINGS.
C. STEEL BUSH RINGS.
D. LANTERN BUSH
E. GLAND RING.
F. GLAND NUT.
G. OIL INLET PASSAGE.
H. PACKING RING.
M GLAND COMPRESSING BUSH.
NECESSARY TOOLS FOR INSERTING & WITHDRAWING PACKING.
LENGTH TO SUIT COMPRESSOR GLAND
LENGTH TO BE SUCH AS TO ENGAGE WITH INNER RING OF LANTERN BUSH
TO SUIT DIA OF COMPR ROD MINUS A SLIGHT CLEARANCE
SPECIAL BOLT & WASHER PLATE FOR WITHDRAWING PACKING COMPRESSORS 3" BORE & OVER.
M GLAND COMPRESSING BUSH, FOR USE WHEN INSERTING PACKING.
EYE-BOLT
PACKING RING H FOR INSERTION WHEN PACKING HAS BEEN COMPRESSED BY USE.
LENGTH TO SUIT COMPRESSOR GLAND
LENGTH TO SUIT COMPRESSOR & SHOULD JUST PROJECT WHEN PACKING IS WITHDRAWN.
LENGTH TO BE SUCH AS TO ENGAGE WITH INNER RING OF LANTERN BUSH
TO SUIT DIA OF COMPR ROD MINUS A SLIGHT CLEARANCE
DIAMETER TO JUST CLEAR SEAT OF NECK RING
TWO FLATS FOR SPANNER
SPECIAL BOLT & WASHER PLATE FOR WITHDRAWING PACKING. COMPRESSORS UNDER 3" BORE.

FIG. 426.—Hall's Metallic Packing for CO_2 Compressor Gland.

of metallic packing shown in Fig. 426, with its accompanying reference to the various parts and also the necessary tools for inserting and withdrawing the packing. The packing consists of alternate rings of steel and soft metal packing with a recessed ring or neck ring *A* at the inner end and a lantern bush, inside which a pressure of oil is maintained by the automatic pressure lubricator. The steel rings are made with different angles, the most acute being fitted at the inner end and the most obtuse at the outer end, which is thus the last one to be put in

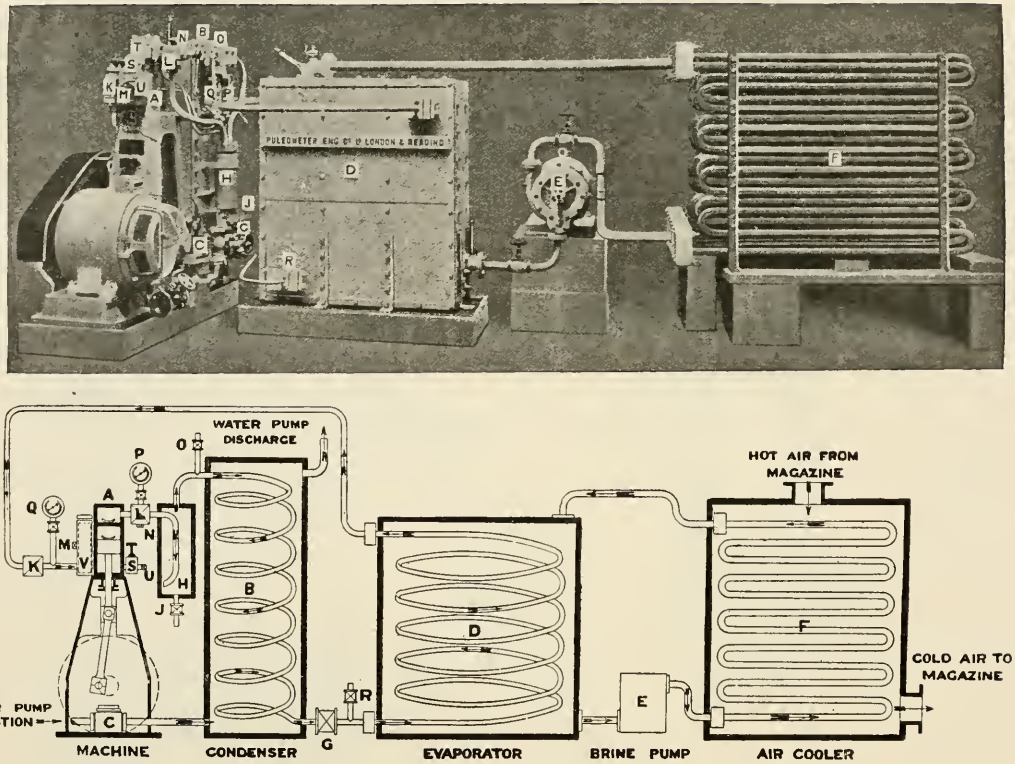


FIG. 427.—CO_2 System for Cooling Magazines, etc.

Fig. 427 shows an elevation and a diagram of a CO_2 refrigerating plant constructed by the Pulsometer Engineering Company for cooling magazines, etc. The apparatus may be driven either electrically, as shown, or by steam or other power. The general principle has already been described, and the parts can be followed from the following:—

Reference to Fig. 427

A, The compressor which compresses the CO_2 *gas*.
B, The condenser, in which, in conjunction with the compression already obtained, the gas is converted to a liquid by cooling.
C, The circulating pump used for cooling the condenser coil.
D, The evaporator in which the liquid CO_2 is evaporated into a gas (at about 28 atmospheres' pressure). The heat required for the conversion of the liquid into a gas is supplied by the *brine*, which is cooled in consequence to a lower temperature. (When in full work the temperature of brine on entry is 40° F.; and on leaving, 38° F.)
E, The brine pump for circulating the brine (weight = 11 lb. per gallon) continuously through the evaporator *D* and air-cooling coils *F*.
F, Air-cooling coils.
G, Regulating valve.
H, Receiver and oil separator.
J, Oil drain from separator.
K, *L*, Stop valves.
M, *N*, Plugs for clearing purposes.
O, Foul gas outlet.
P, Condenser pressure gauge.
Q, Evaporator pressure gauge.
R, Charging valve.
S, Lubricator.
T, Filling plug for *S*.
U, Lubricator valve.
V, Gauge strainer in suction block.

Full instructions are issued with the apparatus, and these should be carefully followed until experience is gained in the actual working.

place. The special metal rings should be a good fit on the rod, and care should be taken that the rod and packing are clean and entirely free from grit. The rod itself must be in a highly polished state to obtain a good efficiency. The screwed gland should not be tightened up too tightly, and on the first day it will require tightening up lightly several times.

The oil used for the gland should be according to the following specification, which is easily obtainable :

Specific Gravity	0·894
Viscosity	135-140 seconds.
Flash point	320° F.
Freezing point	−30° F. or −34° C.

Sea-water should not be used for making brine, but common salt may be used instead of calcium chloride, and it should then be mixed in the ratio of 100 lb. of salt to 1 lb. of caustic soda. The density of the brine should be about 45° Twaddell, or 1·225, which corresponds to a weight of $12\frac{1}{4}$ lb. per imperial gallon.

The gauges show the pressure in atmospheres on the outer circle and the corresponding temperatures on the inner circle.

The compressor is cold, or covered with snow under normal working conditions of temperature and climate, and the delivery pipe should be rather warmer than the hand can bear, but under tropical conditions the entire compressor will become warm and the delivery pipes will be hot. These conditions are principally governed by the temperature of the sea-water, which in hot climates is 80° F. or above (27° C.).

In modern war vessels the Refrigerating plant is almost invariably electrically driven, and this enables the machine to be placed close to the magazine or cooling chamber without undue heating of the surrounding spaces. For merchant vessels both steam and electrically driven systems are used, depending primarily on the type of vessel and its general purposes. The mercantile steam driven sets are usually arranged horizontally, and are a compact unit when the condenser and usually the evaporator are incorporated and installed inside the bedplate.

For warships, where space is always limited, usually vertical machines are fitted.

The condenser coils are made of copper, the compressor of bronze, and the brine-cooling coils are made of iron and fitted in a wrought-iron casing.

The Compressed-air System.—When either the ammonia or the carbonic anhydride is used as the cooling medium, a supply must be carried in the ship, generally in sealed cylinders under compression. Warships are frequently for long periods away from a base of supply; and the compressed-air system has been fitted in several ships, but the CO_2 system is now usually adopted.

The air is not liquefied, as the gas is in the other systems, and the reduction of temperature is obtained by conversion of heat into mechanical work. The air is first compressed to about 10 atmospheres,

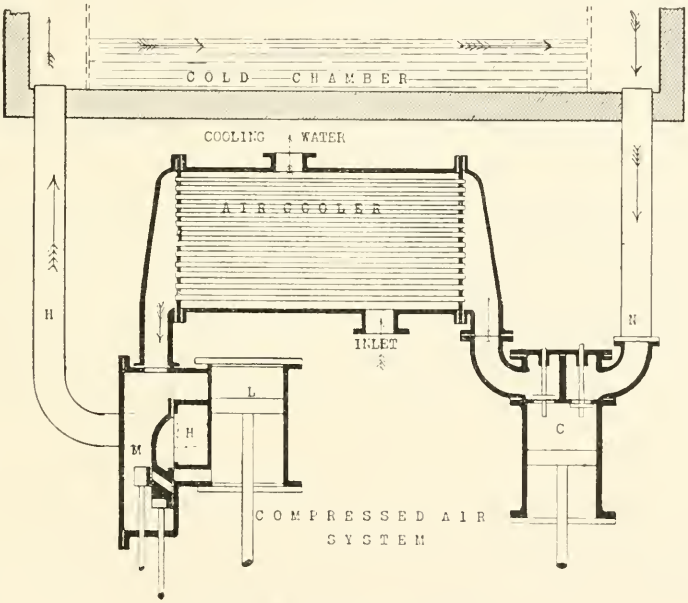


FIG. 428.—Air Compression Refrigerator.

and the compression increases its temperature (as with all gases), and then passes into an air-cooler. The compression cylinder *C* is sometimes fitted with a water jacket. The air-cooler is very similar to a surface condenser: the air passes through the tubes and the cooling water around them. From the cooler, as shown in Fig. 428, the compressed air passes into the slide chest *M* of an expansion cylinder, *L*, where it assists the steam engine in doing the work of compression. A large quantity of the heat remaining in the air is converted into work, and the temperature is therefore reduced considerably, generally to about $-80°$ Fahr. At this temperature and at about atmospheric

pressure, it enters the air trunk, at one side near the top of the refrigerating chamber; the air escapes through holes cut in the trunk, and, being cold, is heavy and sinks.[1] As it circulates downwards and over the parts cooled, the temperature rises and promotes an upward circulation into a similarly constructed return trunk on the opposite side of the chamber. From the return trunk the air is again drawn into the compressor at about a temperature of 20° Fahr., and a new cycle of operations begins.

Drying pipes are sometimes fitted between the cooler and expander; the air passes through pipes which are surrounded by the return air, the compressed air is cooled, and any moisture deposited. Any slight accumulation of water can be discharged through a valve fitted for the purpose. The air trunks for a large installation are about 2 feet square.

There are many points in connection with the practical working of refrigerators which are of great importance, and these are generally noted in the instructions issued by the makers. In each system the success of refrigeration depends on certain simple laws of conversion and work. If the temperature of the incoming gas or air is high, a correspondingly high pressure must be employed, and, in the case of air, a very great ratio of expansion. If the circulating water supplied to the cooler is warm, as in the Tropics, a correspondingly high pressure must be obtained in the compressor. Therefore, in practice, success is fairly certain if the necessary high pressures are obtainable with the plant.

In the compressed-air system the temperature of the air supplied to the compressor is of the first importance, and if, as in some cases, the air is taken from the surrounding atmosphere in a very warm compartment, means must be devised to reduce the temperature of the compartment. Generally some circuit of trunks can be built so that the air travels in a continuous stream through successive cycles of operations, and by this means a low temperature can be obtained after working for a moderate time. In ordinary mercantile practice it is generally necessary to stop the air-compressor engine about every four to eight hours to clear the ice from the slide chest of the expanding cylinder, and to allow some portion of the ice, which is not detachable, to thaw. The slide-chest cover is generally made for quick and easy removal.

[1] 1 cubic foot of dry air at 32° F. weighs 0·08073 lb.
1 ,, ,, ,, 20° F. ,, 0·08274 ,,
1 ,, ,, ,, −80° F. ,, 0·10446 ,,
Thus, air at −80° F. weighs 1¼ times that at 20° F.

PART X

CARE AND MANAGEMENT

CHAPTER XXXIV

THE ENGINEER OF THE WATCH

In the **organisation** of the engineering staff three officers [1] are principally concerned—viz. (1) the Chief Engineer, (2) the Second Engineer, and (3) the Engineer of the Watch. The relative positions and responsibilities of these three officers are exactly similar to those of the captain, the second in command, and the officer of the watch on deck; but with the material difference, that the "Deck officer" can be relieved of his duties and responsibilities almost instantly, whereas the "Engineer of the watch," who has duties in connection with several water-tight compartments, cannot be properly relieved in many instances in less than ten or fifteen minutes.

The Chief Engineer is responsible for everything in connection with his staff and for the efficient working of the machinery and boilers, and the supply of hydraulic and electric power, as well as all mechanical fittings throughout the ship. In this duty he is assisted by the Second Engineer, who, acting as the Chief's executive officer, details all ranks and ratings *by name* for their various duties, and, when necessary, makes arrangements for meeting the requirements of the second in command of the ship. No matter how well the organisation may be

[1] The general terms, Chief Engineer, Second Engineer, etc., are used as they are applicable to all vessels fitted with steam or other power. In large warships the Chief Engineer is usually an Engineer Commander; the Second Engineer is usually an Engineer Lieutenant, and officially known on board as the Senior Engineer Lieutenant; and the Engineer of the Watch is generally of Engineer Lieutenant's or lower rank. In small vessels, the Engineer of the Watch may be an Engine-Room Artificer (E.R.A.) or a Mechanician; but the actual naval rank does not affect the organisation, which must be the same in all vessels.

ordered by the Chief, its efficiency and harmonious working are thus largely dependent on the tact and ability of the Second Engineer in carrying it into effect. He is thus generally the hardest-worked officer in the ship, especially when the staff is a large one and only partially trained. He should keep his Chief thoroughly acquainted with everything which comes under his notice, and also be particularly careful not to omit any detail connected with the organisation—such as changing the duty or station of any officer or man, reporting men for insubordination, etc.

Neither the Chief, nor the Second, nor any officer should give an order affecting the organisation or routine without immediately acquainting the Engineer of the Watch at sea, or Engineer of the Day in harbour. The Engineer of the Watch (or day's duty, as the case may be) should always remember that he is the Chief's representative, and that being placed in that position of responsibility by his Chief, he virtually represents the Captain, who is responsible nominally for everything that goes on in the ship, although the Chief Engineer relieves him in greater part of the responsibility of the engineering department. The assumption of authority by the Second Engineer or other officer senior to the Engineer of the Watch does not relieve the Engineer of the Watch of his duty or responsibility, unless such senior officer gives him a direct order to that effect ; in which case it becomes the first duty of the senior officer to inform the Chief Engineer immediately of what he has done and his reason for doing it. The King's Regulations distinctly define the responsibility of the Engineer of the Watch (see Arts. 972 and others), and he alone is responsible for carrying out all orders received from the Deck Officer of the Watch (who represents the Captain) or from his Chief Engineer.

The same organisation applies to each separate compartment, which during a watch or working hours is always placed in charge of the senior officer or petty officer detailed for that compartment, who, in consequence, is virtually the central pivot about which the organisation moves in that compartment.

Generally the Second Engineer is responsible for the discipline and cleanliness of the staff and department, and in addition he usually details the men for their various duties from day to day. The order and manner in which defects, adjustments, refits, or examinations are taken in hand are always determined by the Chief, and this points to the absolute necessity of the Second and other officers (who are detailed for certain sections of the department) keeping the Chief

thoroughly informed of everything of importance which comes under their notice.

A Watch and Station Bill is prepared in which every officer and man (whose name should not occur more than once) is detailed for his watch and station. It is also usual to make some routine orders and post them conspicuously in the engine rooms for general guidance and reference. These orders should be such that they can be enforced strictly, and are then a very material assistance in the training of the staff in their several duties. (Orders which cannot be strictly enforced might be conveniently issued as "provisional" or "temporary" orders.)

Training is allied to organisation and is also complementary to it, but it is quite easy for one to be efficient regardless of the other. The organisation details a petty officer or man to a particular watch or station, but by training alone can he become familiar with his duties when on watch or at his particular station. With a properly educated engineer, experience in one ship will generally give him sufficient training to carry out his duties in another; but with the men it is a different matter. Men require training in the actual ship until they become accustomed to the various peculiarities and differences in construction and working; and these remarks generally apply to the junior artificers as well as petty officers and men.

Inspection before Lighting Up.—Before lighting up, the boilers should be inspected to see that the water is at the correct height. On no account should a fire be lighted in any boiler unless the water is actually shown at a safe level in the gauge. For water-tube boilers, a height of about 1 to 2 inches above the bottom of the gauge is sufficient; because the water expands when heated, and, by the time steam is up, about a proper working height is shown. The correct working level is generally just above a half glass. With the larger tank boilers, a rise of about 2 to 4 inches is obtained while raising steam.

The engines should be carefully inspected to see that they are all clear, with the turning gear out and no obstructions in the crank and eccentric pits. All valves should be worked and closed again if necessary, but not jambed hard on their seatings. The lubricators should be inspected to see that they are clear, and worsteds properly fitted.

Warming Pipes.—In putting steam on to any part of the ship, it is very necessary that there should be an outlet for the water which has condensed in the pipe, and the drain cocks should be opened first

to allow this water to escape. If water and steam meet in a pipe, there is always danger from water-hammer. The action which takes place is somewhat as follows: the steam on first meeting the water is condensed, and a partial vacuum is formed near the surface of the water; the vacuum greatly increases the velocity of the incoming steam; the momentum increases with the velocity, and the increased force of impact, in combination with the stress caused by difference of expansion, may fracture the containing pipe.

When putting steam on to a cold pipe, before opening out further, about one minute should be allowed for each yard of pipe to be warmed, from the time when steam can just be heard moving in the pipe.

Internal feed pipes are particularly subject to water-hammer, and are frequently found split, especially if fitted near the working water-level.

Time required for raising Steam.—Whenever it is possible, a standard time should be agreed on between the Captain and Chief Engineer for raising steam; this allows many small repairs to be done which would otherwise be left for another opportunity. With *water-tank boilers*, it is not advisable to raise steam in less than six to eight hours. With *water-tube boilers*, the time in which steam can be raised, *in emergency*, is only limited by the time required to properly warm the engines through. Any undue hurry in the process of warming is likely to cause defects from unequal expansion of the various parts, and the time generally required may be safely put at about three hours.

There are few examinations of machinery which cannot be made in two working days, although repairs may take any time. The disposal of a certain amount of necessary routine work allows the engineer freedom to undertake more responsible repairs when the opportunity offers, and thus a fair standard of notice always contributes to efficiency.

Opening out and warming through Engines.—If the full pressure of steam be admitted to the cylinders or steam pipes before they are properly warmed, there is great danger from unequal expansion and consequent fracture. The engines should be warmed through gradually, and when possible, as soon as fires are lighted, the hot air and steam should be allowed to pass into the steam pipes and engines. With water-tube boilers, it is sometimes necessary to keep the bulkhead or sectional valve closed until a feed pump can be started; otherwise the engines, taking away the steam so quickly, leave the water below the proper working level.

The jackets should always be used for warming engines, and the jacket drains opened as necessary. The cylinder and slide-chest drains should be kept open until steam is emitted to the bilge, and then opened occasionally afterwards. If the drains lead into the condenser, they may be left open until after the ship is under way. The engines should be drained when stopped for only a few minutes, so as to prevent any accumulation of water.

The circulating engine should be warmed up with the main engine, and the inlet and outlet on the condenser opened before the condenser is warmed by the steam which leaks through the engines. If only one condenser is required, see that the working one is connected with the eduction pipe, and that no steam can leak into the idle one. As a precaution, the inlet and outlet for every condenser might be opened, and kept open, while under steam.

The hand reversing gear should be tried while raising steam, to get the men into the way of knowing their stations. Start the circulating engine as soon as there is sufficient steam, usually from 15 to 20 lb. The condenser must be kept clear of condensation water by pumping out, if necessary.

When opening out, start at the condenser and retrace the path of the steam from the boilers, in order as below:

Open drains to cylinders and slide chests; H.P. and. M.P. to bilge until steam is emitted; and the L.P. drain to condenser until under way.

Open drains to jackets and steam to jackets.

Open connections for working auxiliary engines, as required.

Open manœuvring valve, about one-half turn generally.

Work main regulating valve and nearly reclose, leaving it very slightly open, so that it is not likely to jamb from difference of expansion.

Open emergency, or quick shut-off, valve.

Open separator drain. If worked automatically, this may be unnecessary.

Open main, or intermediate shut-off, valve; only a moderate amount, as necessary for speed ordered.

Open cross-connecting valve between engine rooms.

Open bulkhead valves as requisite; a moderate amount for speed ordered.

Open boiler steam valves, and notice that the valve is actually moved back.

Warm through the reversing engine, and as soon as steam rises to about 20 lb., sufficient to work the engine, the link should be run over and left alternately " ahead " and " astern " until the engines become lively. The links can be left over, according to the pressure and readiness of some one to check the engines directly they make one revolution.

While raising steam the capstan (for weighing anchor) and steering engines and gear, telegraphs, gongs, steam syren, and other small details required for getting under way, should be prepared, and tried ready for the time ordered.

The funnel guys should be slacked before lighting up, and inspected soon after getting under way, and readjusted if necessary.

Try Engines.—After setting the reversing gear, all-round type, to run steadily, turn a little steam on the engines. The cranks should begin to move a little after about five minutes of this treatment. Then turn on a little more steam until a revolution is made in each direction, as the link runs over. Then stop the link in the "ahead" and "astern" positions, and get about three revolutions in each direction, and continue until a vacuum of about 15 inches is formed in the condenser.

When trying engines, the auxiliary starting or pass valves should not be used to force any crank over the dead centre. Time should be allowed for the evaporation, or drainage, of the water from the cylinders and slide chests. At other times, when the engines are properly warmed, obey the telegraph promptly, because no harm is then probable from using pass valves.

Standing by.—When sufficiently warmed, steam may be shut off the engines, but not from the jackets. The links should be run over occasionally to clear the engines of water, the drains should be opened as necessary, and a little steam occasionally blown through the starting or pass valves to free them of water and keep the cylinders warm.

Under Way.—As soon as the order is given to proceed at a certain speed of revolution, the telegraph is put at " Half speed ahead." Set the link at the correct cut-off first, and then regulate the number of revolutions and the steam pressure in the jackets which are to be used, and shut off the cylinder drains as necessary. For the jackets in use, shut off the drains and allow the water to accumulate until about half a glass is showing in the water gauge on the jacket-drain vessel, and blow down as necessary to keep it at about this level.

Inspect all lubricators to see they are working properly.

Linking-up.—Full advantage should be taken of linking-up by the main and separate links, to get economy and smooth running. The exact amount of linking-up for a certain speed and power is found by experience, and from careful observation of the indicator diagrams obtained and the corresponding ranges of temperature and coal consumption.

Knock on Bearings.—A knock can sometimes be mitigated by an alteration in the separate linking-up of the various link motions to vary the powers in the several cylinders. Hammering gradually increases the maladjustment, and the knock increases unless it is taken up at once. If the link fails to stop the knock, a little water and soft soap assists in forming a lather to fill the slack space between the rubbing surfaces. A knock can always be localised by turning on the water service to each bearing in turn.

Hot Bearings.—If the bearing is only slightly overheated, and cannot be cooled by the ordinary water service, some powdered black-lead is useful, if it can be got into the bearing through the oilways. For more serious cases, sulphur is used until a new bearing surface is formed. After a bearing has once warmed up there is always a tendency to again heat up, and it is good practice to keep the water service on until an opportunity can be made to refit or readjust it. In using water service, care should be taken not to wash the oil from the bearing. The water service should always be prepared for immediate use before getting under way.

Stop Engines.—First shut off steam, then centre the link.

Reversing Engines.—If the telegraph moves from "Ahead" to "Full speed astern," run the links over to "Astern," and then regulate the steam; which generally means opening the valve wider until sternway is got on the ship. All naval engines will stand reversing at full speed without shutting off steam; but the shock of reversal produces leaks and defects, and it should only be carried out in this way at low powers or in cases of apparent urgency.

If the telegraph moves from "Ahead" to "Slow" or "Half speed astern," ease the steam, and then run the links over; if the engines do not at once reverse run the links over again with a little more steam turned on or use the pass valves correctly.

Turbine Machinery

The remarks given above generally apply to reciprocating engines,

but in many respects they are equally applicable to turbine machinery, which is now becoming the most frequent in use and is likely to take the place of reciprocating machinery for practically all purposes in the near future, unless some development brings the internal combustion engine into more general use.

Turbines differ considerably in internal construction but they all follow the Parsons turbine, which was the first to be extensively used for ship propulsion, in the general practice of working and running. The ordinary practice with regard to Parsons turbines has therefore been taken as typical and is as follows:

I. Preparation for Steaming

1. *Forced Lubrication System*

1. The condition of the oil should be examined by drawing off a little of it from the bottom of the tank. Any thick oil should be removed and fresh should be supplied. The oil wells under the bearings also act as settling tanks and should be cleaned out periodically. The strainers in the oil system should also be examined and cleaned frequently.

2. Drain or pump out all *water* from the oil drain tanks; and repeat this operation frequently to prevent the accumulation of water beyond the height of the oil suction.

3. The oil drain tanks should be sounded before starting the oil pump, and again sounded after the system is charged to ensure sufficiency in the tanks, which should be replenished if necessary from the reserve tanks.

4. Open all the necessary supply and delivery cocks and valves on the forced lubrication system; these are usually left open at all times, but inspection should on no account be omitted.

5. Before moving any turbine by steam the oil pump should be started and a free circulation obtained through all the bearings.

6. Notice that the oil flow is directed through the oil cooler, if one is fitted, and that the circulating water system is in order. If an independent water system pump is fitted it should be tried under steam; but after being tried and found correct it should be stopped, otherwise it tends to cool, and prevents the uniform warming of, the turbines.

7. Test cocks, sight holes, thermometers, and pressure gauges are

now usually fitted, and they should be consulted frequently to ascertain the proper flow of oil through each bearing.

2. *Turning Gear, Steam and Drain Connections, etc.*

1. Take out turning gear and secure it, and see that everything is clear. (In some cases special steam connections are fitted for warming through, and the turning gear may be left in ; but it is preferable to first take out the turning gear.)

2. Open all drain valves and cocks in connection with all turbines, except the air pump drain to bilge which should be closed.

3. All self-closing valves between the respective turbine cylinders should be open, unless they are kept closed by springs, in which case they should be free to open.

4. All main, cruising, and manœuvring steam *regulating* valves should be closed.

II. Warming Through

1. *In the absence of Auxiliary Steam*

The hot vapour and steam from the boilers may be allowed to pass through *all* the turbines until a steam pressure begins to show in the boiler gauges ; but the regulating valves should then be almost closed so as to allow only a slight pressure to show in each turbine. (The pressure should not be sufficient to rotate the turbine.) The steam to the shaft gland system of all turbines should be opened for warming through, and the circulating pumps and air pumps should be started as soon as sufficient steam is available.

2. *When Auxiliary Steam is available*

1. The circulating pumps and air pumps should be started before admitting any steam to the turbines.

2. The self-closing and regulating valves, and the various warming-through valves fitted in special cases, should be opened very slightly so as to allow the steam to flow through the series of both ahead and astern turbines. The steam to the gland system of all turbines should also be opened slightly at the same time to warm the ends of the cylinders and rotors ; this system is always kept in operation whether the turbine is actually working or not.

3. In each case steam should be admitted first to the highest pressure turbine of the series, so that it flows through, and warms the cylinders and rotors uniformly in natural sequence, to the condenser. In several instances uniform heating cannot be effected by the above method, and it is necessary to assist the flow by admitting auxiliary exhaust steam to the 2nd expansion of the H.P. main turbine while keeping the L.P. receiver self-closing valve open. The flow of steam should never be great enough to revolve the rotors.

4. *In case of urgency*, each of the series having a direct steam connection may be warmed through independently by easing their respective regulating valves slightly off their faces. Where slide, piston, or other types of manœuvring (differential reversing or working) valves are fitted for supplying steam to the L.P. and astern turbines, the direction of flow should be frequently reversed.

5. In some vessels a small bye-pass valve is fitted for warming through the astern turbines while the turbines are running ahead. This is fitted for preparation for entering or leaving harbour or other times when reversing is anticipated.

III. Try Engines

After the turbines are sufficiently warmed and the gauges applied at the sliding feet show that the cylinders have expanded in a fore-and-aft direction to agree with the amounts measured on the original trials, the rotors should be given a slight movement under steam of about one-quarter revolution, and this operation repeated several times while continuing to warm through before trying engines, and obtaining a few revolutions under steam, ahead and astern.

IV. Leaving Harbour, etc.

Note.—The cruising and other turbines are allowed to revolve, but only those connected with the manœuvring valves are worked under steam.

1. All the necessary auxiliaries, oil, air, and circulating pumps, should be working satisfactorily.

2. All self-closing valves (where fitted) between turbines should be on their seats (closed), but all drain valves should be kept open until the ship is well under way, and steam can be admitted to the main " ahead " series of turbines.

3. The master steam valves should be open to the full necessary extent, and the ahead and the astern manœuvring (working) valves closed, except when actually required for working.

4. The steam pressure in the gland system of the respective turbines should be regulated at each change in the number of revolutions.

V. Under Way

With Main "Ahead" Turbines only working

1. Main steam should be admitted before the steam to the "ahead" and the "astern" manœuvring (working) valve is shut off, and the working valve brought to mid-position (where so fitted).

2. The drain cock on the bottom of the main H.P. regulating steam valve should be opened to drain off any water, and thus allow only dry steam to enter the cylinder.

3. All drains to cruising and separate astern cylinders (not in use) should be left open to prevent accumulation of water and to ensure a vacuum in those cylinders.

4. Self-closing valves on cruising and other turbines (not in use) should be closed, and those on the working cylinders opened.

5. Steam may then be admitted gradually to the main H.P. turbine until the required speed is attained.

VI. Changing from Lower to Higher Expansion Turbine

1. Warm through Higher Expansion turbine.

2. Screw back self-closing valve on Lower Expansion turbine so as to leave it free to open.

3. *Open* steam to Higher Expansion turbine and at the same time *close* steam valve to Lower Expansion turbine.

4. Adjust steam pressure for revolutions.

5. Close drains on Higher Expansion turbines, and adjust gland pressures.

VII. Changing from Higher to Lower Expansion Turbine

1. Open drains on Higher Expansion turbine.

2. Close the self-closing valve on Lower Expansion turbine as far as possible.

3. *Open* steam to Lower Expansion turbine and *close* steam valve to Higher Expansion turbine, at the same time assisting self-closing valve to close.

4. Adjust steam pressure for revolutions, and adjust gland pressures.

CHAPTER XXXV

BOILER PRESERVATION AND REPAIRS

Internal Corrosion.—The principal causes of internal corrosion and decay of boilers are evolution of oxygen, air, and gases from the feed water ; keeping the boilers when not in use only partially filled with water ; wetness or moisture in a boiler supposed to be empty ; dirty condition of the heating surfaces (internally) when under steam ; and galvanic action originating between different materials used in their construction.

The general treatment of the internal parts is the same for every type of boiler ; but for some boilers the detail is more extensive, and greater care is required because the structural conditions adversely affect the circulation of water and steam. Sea-water contains more air and gas than shore-water. Distilled water contains less than shore-water ; and, because it is desirable to exclude air and gases from the feed water as far as practicable, distilled water should be used at all times.

Air, oxygen, and gases are expelled from water by raising its temperature, and therefore hot feed water is preferable to cold ; and when either sea or shore water is used for filling the boilers it should be immediately warmed to expel the air and gases, and the air cock or the safety valves should be opened to allow the air to escape before the boiler is shut off for preservation. The boiler should never be allowed to remain open for a longer period than is absolutely necessary for cleaning and repair, and if it is not immediately convenient to close it up, a wood fire or an airing stove should be lighted in the furnace to keep the general temperature above that of the surrounding atmosphere, and thus keep the boiler dry and free from moisture both internally and externally.

With water-tube boilers, into which cold feed water can be pumped without injuring the boiler from unequal contraction, the boiler is

usually filled to the crown and 5 to 10 lb. pressure obtained in it immediately after the fires are out (by using the auxiliary service or the hand pump), and, the water being warmed on entry, the air and gases should be allowed to escape freely before closing up the boiler for preservation. With cylindrical boilers, the addition of a quantity of cold feed water while the boiler is still warm produces racking strains, and therefore the boiler must be allowed to cool slowly ; generally it is sufficient to shut the boiler off immediately after the fires are out, and to pump in only a sufficient quantity of water to raise the level to a few inches above the working height, say to a full gauge glass. If it be decided to fill the boiler to the crown, the water should be warmed, and the air, etc. allowed to escape before the boiler is permanently closed for preservation.

Internal corrosion usually takes place in the region of the water-line (working level) or parts of a boiler where the nascent oxygen freed from the water strikes the interior surfaces.

Corrosive action is checked in boilers by elimination of air and by fitting pieces of rolled zinc, about 1 foot long, $\frac{1}{4}$-inch thick, and 6 inches wide, in selected positions ; and zincs, enclosed in galvanised wire cages, should be placed just *above* the working water-level. These precautions, together with others which may recommend themselves in particular cases, and the abandonment of the use of soda, have been found most effective in arresting corrosion.

Galvanic action, originating between different materials used in the construction of the boiler, is checked by suspending *zinc slabs* in electrical contact with various parts of the boiler. The zinc is attacked by attracting all galvanic action to itself until it is eaten away, or electrical contact is broken, thus protecting the boiler material. Zinc also counteracts corrosion by corroding away—this is the usual action above the working level,—and it is thus somewhat difficult to determine whether either galvanic action or corrosion causes pitting of the interior surfaces. In any case the careful fitting of zinc slabs acts in a manner beneficial to the life of the boiler material, and additional zinc protectors fitted near pitting already observed frequently arrest further action.

Lime is mixed with the feed water to prevent any acidity—which assists galvanic action—of the water attacking the internal surfaces of the boiler, and as a nucleus for collecting particles of grease. A tank is fitted in the engine room, in connection with the main feed tank, in which a certain proportion of lime—roughly, from 1 lb. of lime for

each 1 ton of distilled "make-up" feed to 5 lb. of lime for each 1 ton of sea-water "make-up" feed—is mixed with a small quantity of water taken from the main feed supply. The clean lime water is allowed to gravitate from the upper part of the lime tank into the feed tank, where it mixes with the feed water and is pumped with it into the boiler. The quantity of lime used should give the water sufficient alkalinity to turn the natural colour of litmus to *blue*. (See Chapter IX., Chemical Tests.) The condensed water, the feed water, and the water in the boilers are tested for acidity at least once daily, and the lime should be added in small quantities at regular intervals.

A *lime strainer* is fitted on the outlet pipe in connection with each economiser or hot water collector to prevent the passage into the generator feed pipe of particles of dirt or grease collected by the lime. The strainer should be removed and cleaned on each occasion of cleaning the boiler interior.

Soda should never be used in any boiler under ordinary conditions of working. In addition to contributing to the formation of acid (hydrochloric acid particularly), it makes a priming mixture which is very difficult to overcome. Soda in a concentrated form is, however, sometimes mixed with the boiler water and boiled continuously so as to clean the interior of the generating tubes and other parts. The usefulness of this method of cleaning is by no means established, and, with proper filtration of the feed water, should be unnecessary. After using soda in the boiler, it should be carefully washed out with fresh water; or, if a sufficient supply of fresh water is not available, sea-water may be used for the earlier and rougher part of the operation of washing out.

Periodical Cleaning.—A great *loss of evaporative power* (and fall in economy of fuel) ensues from dirty heating surfaces, and allowance must be made for ready access to clean them periodically. On ordinary service, when clean fresh water only is used for feeding the boilers, and the grease extractors are efficient, every boiler requires a complete internal examination and cleaning at least once in six months; and no boiler should be under steam without examination for more than a total period of thirty days. At such times, every door should be removed and every tube searched and cleaned as far as practicable; the zinc protectors refitted or renewed, and electrical contacts ensured by cleaning and refitting them; and the feed and other valves and the automatic feed regulators cleaned and refitted.

After the internal cleaning of a water-tube boiler is complete, it

should be washed through with clean fresh water ; then closed up and tested by water to at least the working pressure so as to ensure that all joints are absolutely tight. This test is, of course, supplementary to the periodical water-pressure test, and is usually made after any minor repairs have been effected, or making good leaky joints.

When dirty or greasy or brackish feed water is used, the examination and cleaning must be made more frequently ; and with cylindrical boilers, particularly those working under forced draught and high pressures and temperatures, the necessity is still more urgent.

The *exterior*, or fire side *of the heating surfaces*, particularly the tubes, require cleaning or sweeping when under steam once in twenty-four hours (according to the rate of combustion and class of fuel used).

After steam is down, the external parts of the heating surfaces, etc., require a thorough cleaning with steel wire brushes or other manual appliances, and care should be taken that all dirt, soot, etc., are removed from behind the elements, flame plates, and brickwork, because when under steam the natural expansion of the material of the boiler may otherwise be restricted and thus bring unnecessary stress on the casings and supports ; the firing of soot behind these parts also tends to rot and burn away the more permanent parts of the boiler casings.

Lack of facility for manual cleaning of the sooty surfaces of the tubes and heating surfaces of the boiler in a great degree accounts for the considerable falling off in efficiency and in economy of fuel after fires have been alight only a few hours. Further remarks on this and kindred subjects will be found with the description of the various types of boilers.

It is important that the air casings should be sufficiently tight to prevent the entry of cold air at any point above the lowest part of the heating surface of water-tube boilers ; such leakage has been a frequent source of loss of evaporative power, and a cause of smoke formation.

The rate of transmission of heat in b.t. units per second per one square foot of surface of plating, or heating surface (approximately)

$$= K \times \sqrt{\text{Difference in temperature at opposite faces of plate.}}$$

For steel, $K = 0{\cdot}07$ to $0{\cdot}08$, when the plates are clean, and is, within the limits of ordinary boiler practice, independent of the thickness of the plate.

When the water surface of the plate is dirty and covered with lime, or greasy scale, the difference of temperature between this side of the plate and the water rapidly increases if the rate of evaporation remains constant. With about $\frac{3}{16}$-inch of greasy scale and with high rates of combustion the difference becomes nearly 400° Fahr.

With about $\frac{1}{16}$-inch of clean lime scale, nearly free from grease, the fuel consumption increases by about 15 per cent: the consumption increases almost in direct proportion to the thickness of scale. A loss in the rate of transmission of heat from the presence of scale considerably affects the safe generation of steam at high powers, and has a distinct bearing on the absence of sufficient allowance for unequal expansion.

External Examination.—The principal causes of external defects developing in boilers are: unequal expansion; excessive forcing; inadequate repair; moisture on the surfaces exposed to corrosive action; and galvanic action caused by leakage from electric circuits.

Unequal expansion of cylindrical boilers after the fires are lighted is always provided for, as far as possible, in the design, because the boiler must undergo some considerable change of form when one part becomes hotter than another. Thus the top of the boiler which is in contact with the steam may have a temperature of about 380° Fahr.; while the bottom, which is exposed to a cold air current and cold water, may be only at 70° to 100° Fahr. It is this difference of temperature, and consequent unequal expansion, which causes practically all the unavoidable defects in boilers. The radius of curvature of the end plates where they are dished to meet the shell plates is made fairly large because the shell tends to expand more than the end plates, being subjected to a greater average temperature. Similarly, as shown in Fig. 26, other parts are dished with large radii of curvature. Attention has already been called to the expansion in length of the furnaces, and the allowance given to the portable and other stays in proximity to them; to the now usual method of fitting bridge, dog, or girder stays in place of vertical stays above the combustion chamber; to the absence of screwed stays below the combustion chambers; and to the flexibility of the furnaces themselves. Too great care cannot be given to these matters, but, unfortunately, they are among those which the draughtsman frequently leaves unconsidered.

An *allowance for unequal expansion* is also necessary to all parts of a water-tube boiler. The generating tubes of the Niclausse and of the Dürr boiler are only rigidly connected at one end, and they are

thus free to lengthen ; and therefore the expansion of each tube when clean is entirely independent of any other part of the boiler.

In the Belleville boiler, each element is independent of any other and is only securely held at its connections with the steam chest and with the feed collector ; the tubes can expand moderately freely in length.

In the Babcock and Wilcox boiler, the allowance for expansion is comparatively little, but in the more recent types there is considerable improvement. The greater inclination of the tubes to the horizontal, compared with the Belleville type, promotes a better circulation with a more moderate temperature, so that on the whole the allowance for expansion may be considered sufficient.

In boilers of the upright-tube type, the rate of circulation is very rapid and the allowance for expansion is ample if the tubes have a curvature of about $\frac{1}{4}$-inch or over in each foot of length. Even in a straight-tube Yarrow boiler it is considered that expansion is allowed to some extent by the partial rotation of the water drums on their respective seatings, to which they are not rigidly attached. This does not, however, allow for difference of expansion of tubes in the same row, from front to back, and in all the more recent types of Yarrow boiler the tubes in the rows next the fire are made with some curvature.

As steam is raised, all boilers increase in height. The upper part of the upright-tube type is then supported by the tubes from the water chambers. In boilers of the horizontal-tube type, the steam chest sometimes supports the lower parts by suspension, and the boiler framing is then adapted to supporting the steam chest. Chocks and guys are fitted, as with cylindrical boilers, to prevent movement of the heavier parts of the boiler when in a sea-way.

The straight tubes of water-tube boilers *become bent*, to a more or less extent, whether the tube is held either at one end only or at both ends. This bending is caused by unequal expansion of the fire and uptake faces of the tube, and may be best illustrated by considering a nearly horizontal tube through which mixed steam and water are circulated when steam is being generated. The upper part of the tube is principally filled with steam and the lower part with water ; heat passes comparatively quickly into the water and tends to keep cool the material of the lower part of the tube ; but the upper part, although subjected to a lower heating temperature, is generally at a higher temperature than the lower part because steam is a very poor conductor of heat, and therefore does not convey the heat of the upper

part of the tube either to itself or the water beneath it. The upper part being thus lengthened more than the lower part, produces hogging or bending of the tube upwards away from the fire (Fig. 429).

Roughly, the average temperature of the metal of a clean tube may be taken as about one-third that of the fire or gases near it; but the rate of circulation and rate of combustion also affect this temperature materially, as well as the cleanliness, or otherwise, of the water surface of the tube. For an ordinary tube next the fire, with steam up in the boiler, the expansion in length of a 7-feet tube would probably be about 0·4 inch; this allows for the expansion of the tube from a cold water temperature, say 100°, up to about 800° Fahr.

Bent *tubes of the nearly horizontal-tube type boiler should be straightened* when necessary, and a limited curvature may be given them towards the fire so as to lessen the continual trouble of straightening them. The curvature should in neither case be allowed to exceed that necessary to keep the *exit* opening above the highest point of the curve, and all

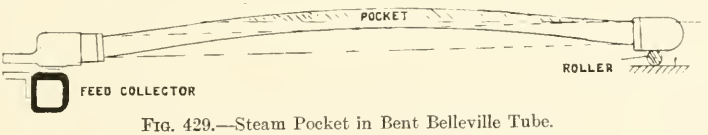


FIG. 429.—Steam Pocket in Bent Belleville Tube.

points in the curve should be above the *inlet* opening; otherwise a steam pocket may be formed (shown shaded in Fig. 429), which tends to produce local overheating of the tube material from insufficient circulation, or from its possible tendency (Belleville type) to reverse the direction of circulation.

Water-tube boilers of the straight-tube type require particular observation for deflection after each period of steaming. The curvature is not of much importance in itself unless the tube is left with a tendency to form steam pockets; this applies particularly to the Belleville boiler where the inclination of the tube is only $2\frac{1}{4}°$ to $2\frac{3}{4}°$ from horizontal, and, as a rule, the curvature should not exceed one inch because of the possible formation of a pocket and the tendency to leakage at the nickel ferrule joint.

In the Dürr and Niclausse boilers the generator (external) tubes bend, and it is advisable to straighten them because the internal (circulating) tube remaining straight comes into metallic contact with the generating tube and interferes with the natural circulation.

In the Babcock and Wilcox and the Yarrow boilers, bending or

change of curvature is not important so long as it is gradually made in the same direction ; but alteration in form tends to fatigue the material, especially when aggravated by alteration of stress. Alteration of curvature occasionally causes leakage at the tube end joints (easily detected when not under steam by presence of scale near the leak), and these should be remedied by lightly expanding the tube into the tube plates.

Generally speaking, the repair of any water-tube boiler, particularly of the parts subject to steam and water pressure, can be done on ordinary service by the ship's staff, but the repairs to the casings are not so easily dealt with. Notwithstanding the brickwork surrounding the furnace, the hot flames and gases, with slight diminution of temperature, may strike the side and back casings and warp and burn them away. This difficulty with casings is common to all boilers of the horizontal-tube type, described in Chapter VIII. Although seldom apparent in the early life of boilers, the defect is a constant anxiety and loss of economy after two or three years' ordinary service. With upright tubes, the casings at the front and back only (which are more easily protected by brickwork) are subject to the hottest flames and gases ; but the side casings are subjected to only a moderate temperature, because the gases have been cooled by passing through the passages formed by the generating tubes. This comparison is strongly in favour of the upright-tube type.

The essential feature of a *boiler casing* is that it should be air-tight, and the *doors* in connection with it should be properly bedded on asbestos cording so as to maintain the general air-tightness of the casing. Leakage tends to support combustion in the region of the leak, and thus the flames burn away or buckle the casings and increase the tendency to leakage. A supply of cold air, other than that immediately above or below the fire, is not conducive to economy.

In some types of water-tube boilers *baffles* are fitted among or near the tubes to deflect the flow of the hot gases, and to distribute their flow equally to communicate their heat to the water in the boiler. It is therefore important that they should be properly placed and in good repair. Baffles are also fitted to check the flow at certain parts of the boiler, between the generating tubes and headers and the casings particularly ; these are usually called *flame plates*, and their special use is to prevent the impact of the hot flames on the casing and thus to protect it. Baffles and flame plates are fitted so that they are easily removable.

The condition of the *brick-work* has an important bearing on the durability of the boiler casings, as well as on the economy of steam generation, and for both reasons it should be kept in the best possible repair.

Instructions for the Better Preservation and Working of Yarrow Water-tube Boilers

(These Instructions, for which the author is indebted to Messrs. Yarrow & Co., appear to be applicable to all modern boilers, and in accordance with the best practice of the present day, and are therefore given with only slight variation from the original.)

Precautions when Lying Up.—For the better preservation of a water-tube boiler when lying up, it is advisable that it should be emptied and drained of water, and thoroughly washed out internally with clean fresh water. Any accumulation of soot or ashes should be removed from the tubes and tubeplates. This is of the utmost importance, because if moisture become absorbed by the dirt which collects on the heating surfaces, corrosion will soon commence, and, when once started, will increase rapidly. The casing should be carefully swept on the inside. After being thoroughly cleaned the outside of the tubes at the lower ends should be sprayed with tar from time to time so as to give them a good protective coating.

Drying Boiler.—A small coke fire should be lit in a suitable portable receptacle, which may be placed in the furnace to thoroughly dry both the inside and the outside of the boiler, the coke fire to be kept far enough from the tubes to avoid overheating them; in the case of a boiler burning coal the coke fire should be placed in the ashpan and a few fire-bars removed. The man-hole, mud-hole and casing doors being off, the vapour formed will escape.

A paraffin flare lamp or electric heater is preferable to a coke fire, as the sulphur from coke causes corrosion unless care is taken to obtain coke free from sulphur.

Internal Examination.—A governing feature in the design of the Yarrow boiler is the facility for examination of every portion internally, and this can be effected from the steam drum and water drums, access to which is gained by simply removing the man-hole doors.

When the boiler is dry, a brush or other appliance should be passed through each tube. Every tube in the Yarrow boiler can be examined and cleaned throughout its entire length. The examination can best

be done by holding a light in the lower drum and looking through the tubes from the upper drum or by passing an electric finger up the tubes. By this means, any obstruction or serious corrosion becomes visible, and should scale or obstruction be found to exist, it should be at once removed. This offers no serious difficulty because the tubes are usually straight.

Preservation when Lying Up.—If the boiler is intended to lie up for a lengthened period, some quicklime in suitable trays should be put into the lower and upper drums. The drums should then be closed up to exclude the air, care being taken to remove the lime before again filling the boiler with water. The object of the quicklime is to absorb any moisture that might remain in the interior of the boiler.

Another reliable practice when laying a boiler up is, after it has been thoroughly washed out, to close up all man-hole and mud-hole doors, and to quite fill the boiler with clean water, adding 1 lb. of lime to each ton of water, this lime being dissolved in the water before it is put into the boiler. Care should be taken before again starting the boiler under steam to thoroughly empty it.

A coke fire or electric heater should be provided, so as to keep the boiler at a temperature slightly in excess of that of the surrounding atmosphere, otherwise moisture may collect on the outside surfaces and cause deterioration. This is particularly important in summer time when the daily variation of temperature is greater and humidity of atmosphere is often sufficient to deposit dew.

Funnel Covers.—The funnel covers should be put on to prevent rain wetting the tubes and casings. The funnel covers should always be fixed when the vessel is not under steam.

Brick-work.—If the boiler is to lie up for a lengthened period it is also very desirable that the brick-work should be removed and only replaced when required.

Raising Steam.—When it is intended to raise steam the boiler should be filled with water to half-way up the gauge glass, and 1 lb. or 2 lb. of ordinary lime per 1000 gallons should be added in the form of milk of lime. Care must be taken that the lime is well mixed before being put in the boiler, and the lime-water should be passed through a fine strainer. If possible steam should be raised slowly when the brick-work is new or recently repaired, but at other times steam may be raised more quickly. See that all the boiler fittings and feed arrangements are in good working order whilst raising steam.

Water Pockets.—In the case of the larger Yarrow boilers access to the lower drums is by means of a manhole, both for cleaning and repairs.

In the case of the smaller boilers, the water pockets can be cleaned by means of mud-holes, but for large repairs a bolted joint is provided. The joint is made with red lead or in an emergency with asbestos metallic or klingerite sheeting $\frac{1}{16}$ of an inch thick. Before breaking this joint, the weight of the boiler should be carried on lugs provided for that purpose at each end of the lower tubeplates. When remaking the joints of these water pockets, after having screwed the joints up as tightly as possible, steam should be raised to 10 lb. per square inch to thoroughly warm the boiler, and the bolts in the joints finally tightened up. It is only contemplated to break these joints in case of important repairs.

Precautions when in Use.—When working, every opportunity should be taken to shut down each boiler in rotation in order to examine the brick-work and other parts of boiler and clean the tubes inside and out along with the remainder of the boiler. The two or three rows of tubes nearest the fire require more careful attention than those which are farther from it. If any accumulation of sediment is found it should be removed before the boiler is started again and the source of the deposit ascertained.

Stoking Boilers burning Coal.—A thin even fire should be kept, taking care to keep the corners of the grate covered. The thickness of the fire is to a great extent determined by the class of fuel used, and the amount of forcing adopted, but in any case flame should not be seen coming from the funnels, as that is an indication that the gases have not been properly burned. On the average a thickness of fire of from 5 to 6 inches has been found, with Welsh coal, to be suitable when working up to half-inch air pressure. When working at 3 inches air pressure the thickness of the fire may be increased to 9 inches. The fire doors should be kept open as little as possible when firing, so as to avoid cooling the furnace. In the best practice the door is opened and shut between each shovelful.

When charging the furnace the coal must be thrown on in the exact places where required, and not piled up at the front end of the grate and afterwards pushed back, as is customary with ordinary marine boilers. With the aid of coloured glasses the firemen can see where the holes in the fire are before they begin firing.

Feed Regulation.—In regulating the feed, the check valve

should be altered very little at one time. Careful adjustment is required at first, and when once set to a suitable area of opening, little further attention is required.

No Oil to be admitted to Boiler.—No oil should be allowed to get into the boiler. If any oil is used for the internal lubrication of the machinery, it should be mineral oil, and generally it is found that oil can be dispensed with altogether, which is very desirable. If reciprocating engines are installed, as little oil as possible should be used for lubricating the piston rods, because a certain amount of oil invariably finds its way into the interior surfaces by this means.

Auxiliary Engines.—Special care should be taken that the auxiliary engines are not of such a character as to involve the use of oil for internal lubrication. The auxiliary engines should be run without any internal lubrication whatever, and if any lubricators are fitted they should be removed.

Feed Filter.—An ample area of feed filtering surface should be provided, and care taken to keep the material used clean.

Water to be Alkaline.—The water used in the boiler should always be pure, and only when unavoidable should it be obtained from a doubtful source on shore, as that will often lead to the formation of scale and corrosion. Tests should be made from time to time to ascertain that there is no acid contained in the water in the boiler, and not only should it be alkaline, but it must be definitely so. For this purpose from 1 lb. to 2 lb. of ordinary lime per 1000 gallons of "make-up" should be pumped daily into the feed as milk of lime, or more if found necessary, to ensure the water being decidedly alkaline. Care should be taken that connections between the boiler and the water-gauges are kept clear from any accumulation of lime.

Sea-water.—No sea-water on any account should be allowed to get into the boiler, and for this reason care must be taken that the condensers are tight, that the evaporator does not prime, and that all sea connections are properly shut. If, however, sea-water does get into the boiler, double the ordinary quantity of lime should be used with the feed, the fires must not be forced, and the density kept as low as practicable.

Ashpit and Fire Doors of Boilers burning Coal.—The ashpit doors must always be kept properly working so that in the event of a boiler tube bursting or steam suddenly escaping through any other cause, it may not find its way into the stokehold. For the same reason the fire doors should be kept closed, except when stoking.

In the event of a serious leakage of steam in the stokehold the fan should be immediately turned on to force the escaping steam up the funnel, the stokehold doors should be closed, and the feed pumps turned on full speed.

When burning oil, the oil fuel pump should be immediately stopped or the oil fuel supply to the damaged boiler shut off.

Casing Joints.—The casing joints and doors must all be perfectly tight, because leakage of air will cause rapid destruction of the casings as well as loss in efficiency, and care must be taken not to allow any large accumulation of ashes or soot in any portion of the casing.

Tube Plugs.—In the event of a tube giving way, the ends should be closed by plugs provided for the purpose. No appreciable reduction of efficiency would be found, even if after lengthened service 10 per cent of the tubes are inoperative.

Accumulation of Gas.—When opening the boilers or any parts connected to them, such as cylinders, pipes, condensers, etc., great care must be taken to prevent any open light being near, as sometimes explosive gases are formed by the zinc in the boilers which may catch fire and cause serious injury unless every part is well ventilated.

CHAPTER XXXVI

CARE AND ADJUSTMENT OF MACHINERY

The main propelling machinery is always erected in the workshops before its final erection in the ship. By this system the alignment and proper fitting of all parts are ascertained, and any discrepancy remedied while still in the building stage, and usually a water pressure test is made of the cylinders, jackets, and other fittings in the shop. Although this preliminary building up of the engines, only to be taken down and rebuilt, seems somewhat of a useless undertaking, experience shows that a large saving in the cost of labour and better fitting and adjustment are obtained. Before the machinery is erected in the ship it is necessary to bore the stern tube and other bearings, and to make certain that the axis of the shaft, including the crank shaft, lies truly in line. The successful working of the machinery is largely dependent on this accuracy, and very careful sighting arrangements are made for this object alone. This part of the work is usually outside that of the sea-going engineer, and can only be done with sufficient accuracy by using the proper instruments. The sighting is preferably done at night so that there shall be no confused light, and for the same reason it is usual to fit a screen over the apparatus fitted outside the ship. The sighting apparatus consists of a small hole made to exactly coincide with the axis of the foremost bearing, and behind it (at the forward end) an electric lamp of 8 or 16 candle-power is fixed. In each bearing a similar hole, but adjustable both horizontally and vertically, is set, and there is a fixed hole at the aftermost end (stern bracket bearing) which is fitted with an eye-piece. By careful adjustment, using cross-lined sights if necessary, each hole is brought in line between the foremost and after holes until the light shows clearly from end to end. From these holes the bearing circles can be struck out, and the parts bored accordingly.

Alignment of Shafting.—After the shafting has been fitted in

the ship, the alignment can be tested at any convenient time by disconnecting the couplings and drawing the shaft clear from each in turn. The amount of drop, or variation from parallelism of the coupling flanges, can be measured by using feelers, that is, thin sheets of steel graduated for the purpose in one-thousandths of an inch. If the measurement shows any appreciable wear of the bottom brass, it should be removed and remetalled, so as to bring it up fair with the remainder. A common method of gauging is shown in Fig. 430, and the drop is measured by inserting a feeler between the gauge and the top of the shaft. The diameter of the shaft must also be gauged to obtain an accurate measurement. If remetalling is impossible, the brass may be raised temporarily by means of a liner of the correct thickness placed below it; but this method requires great care and accuracy, and is not generally recommended because any discrepancy in fitting is liable to bring local stress on the brass and tend to distort it. Generally the wear on stern bearings is greater than that at the crank end of the shaft, and consequently there is a droop of the shaft towards the propeller end. If this is general and fairly gradual, no unpleasant consequences need be expected; but if it is excessive, the imperfect alignment brings great alternating stresses on the coupling bolts from the bending of the shaft at every revolution; the frictional resistance is also increased, and there may be a tendency to whipping in very long and small shafts. With large and comparatively short shafts, well supported by bearings at short distances, whipping is not a usual occurrence, and unpleasant experience is seldom occasioned by it.

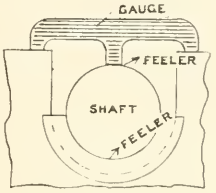


FIG. 430.—Gauging Alignment of Shaft.

Alignment of Pistons, Rods, and Guides.— In *horizontal* engines, the piston rings have a distinct tendency to wear down and allow the body of the piston to come in contact with the liner or working barrel of the cylinder. The alignment can be tested from outside the cylinder by moving the piston to opposite ends of the stroke (*see* Fig. 431 for vertical engine) and carefully measuring the heights A and B of the rod above the guide E in the two opposite positions. This measurement should be made as near the cylinder as possible, from the same point on the guide surface, and shows any difference in wear of the piston rings and guide. If time permits, it is preferable to open the cylinder and gauge the actual position of the piston with respect to

the barrel, and line it up if necessary by adjusting the thickness of the cod piece between the packing ring and body of the piston. When making this adjustment it is usual to set the piston rather above the central position, to allow for subsequent wearing down; this amount should not exceed $\frac{20}{1000}$ths of an inch, or trouble may be experienced with the rod packing from inaccurate alignment of the piston and guide. This alignment should be tested after centering the piston, and if necessary the guide should be lined up also.

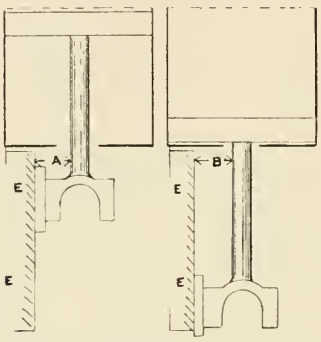


Fig. 431.
Alignment of Piston, Rod, and Guide.

In *vertical engines* the pistons seldom require realignment, because the wear is approximately equal all round. The procedure is the same as for horizontal engines, but it is usually the guide that requires attention. With heavy pistons without tail guide rods it is good practice to fit solid blocks in lieu of three or four of the spiral springs in the piston immediately above the guide, and thus prevent oscillation and knocking. These blocks should be carefully fitted so as not to work loose vertically or sideways; mysterious knockings have been stopped in this way which were originally attributed to water-hammer or slackness in the guide and crosshead bearing.

The piston packing rings in both horizontal and vertical engines should be allowed sufficient freedom of movement, without leakage, between the flange of the piston and the junk ring. A good practical method of testing this adjustment is to insert, before screwing up the junk ring, slips of thin tissue (cigarette) paper at intervals between the packing and junk rings. When the junk ring is screwed hard up, the slips should be just withdrawable without tearing the paper. If measured by micrometer or soft lead wire, the clearance should not be less than $\frac{2}{1000}$ths of an inch for cast-iron rings of any ordinary width, and of $\frac{1.5}{1000}$ths *per inch* of width for gun metal or phosphor bronze.

When the crosshead bearing is disconnected from the piston rod and slipper, it is quite possible for the latter to appear out of alignment with the guide. The slipper always has some clearance sideways, or lateral play, and it is therefore possible for the rod to appear to be out of line. Therefore, this alignment should be tested by applying

a set square in the top brass (Fig. 432), and the tongue of the set square should then be parallel to the rod ; in making this measurement care should be taken that the rod is parallel, *i.e.* of exactly the same diameter between the points of measurement. If this alignment is apparently correct, *A* and *B* being equal, then connect up the rod and top bearing, and turn engines with the weight of the piston, etc., resting on the top half of the crank bearing. If the alignment is correct, the crank end of the connecting rod will have no movement endways which is not shared by the top end bearing. While testing this alignment, the crank shaft should be prevented from axial movement by nipping one of the thrust-shaft collars fairly tightly between two thrust rings, or by actual observation of any axial movement.

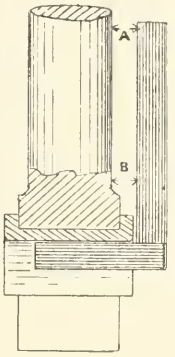


FIG. 432.—Alignment of Connecting-rod Brass.

The *alignment of the connecting rod* is of great importance to efficient working, and should be carefully tested when making adjustments or re-fitting the connecting-rod bearings. The liners between the top and bottom brasses should be removed from each end in turn, and the brasses tightened up until the rod is just movable by hand ; the opposite end is then disconnected, and the rod should swing freely and exactly over its pin. As pointed out above, the guide may be canted sideways (*i.e.* fore and aft), and throw the disconnected end out of alignment ; in this case, which is common with heavy engines, the bearings at each end should be tightened up fairly closely (about midway between " hard up " and the usual " working clearance "), and the " marking " (red lead) will clearly show the hard places after giving the shaft a complete revolution. If necessary, the hard places should be eased away from each bearing in turn until the pins bed fairly in the metal at top and bottom for a vertical engine, or front and back for a horizontal engine.

Adjustment of Connecting-rod Bearings.—Although it is not necessary to test the alignment so frequently as the adjustment, it should be remembered that no adjustment can be entirely satisfactory unless the alignment is good. In each bearing, the pin should bear evenly on the metal of each of the two brasses for a length around the circumference of the pin equal to the diameter of the pin. This leaves $\frac{1}{6}$th of the circumference at each lip of the brass clear of the

metal at and near the positions of the liners, as shown in Fig. 433. The thrust and pull exerted through the connecting rod is in the direction of its length, and therefore clearance at the liners reduces frictional resistance without decreasing the effective bearing surface.

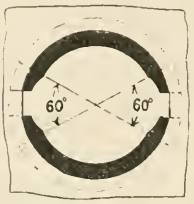


Fig. 433.
Clearance in Bearings.

The adjustment is tested by using *soft lead* wire. Four pieces of wire are generally used for large engines; one piece is laid half round each end of the pin just clear of the oil-containing strips, and one piece fore and aft about one or two inches away from each side of the oilway in the crown of the brass. The bearing is next tightened up to the proposed setting, and then slacked back until the wires can be removed. The thicknesses of the wires are carefully measured by a micrometer gauge and then compared with the usual requirements, which in the Navy are usually recorded in a "Lead book" as below. (Specimen.)

S.M.P. Crank Bearing

Date.	Marks on Nuts.			Thickness of Leads.		
	Found at	Hard up	Left at		Crown	Horns
8/8/07 Port nut	6 on $1\frac{1}{2}$	6 on $3\frac{3}{4}$	6 on 1	Forward	25	40
8/8/07 Starboard nut	6 on $1\frac{3}{4}$	6 on 4	6 on $1\frac{1}{4}$	Aft	25	40

Remarks.—Bearing refitted and eased away at horns.
Initials of officer superintending adjustment—K. C. B.

The liners should be of the correct thickness, so that when the bearing is tightened up to the correct adjustment they should be firmly nipped between the butts of the pair of brasses. Before running at full power, the bearings of large engines are usually slacked so as to give about 2 to 3 thousandths of an inch greater clearance. After the bearings are adjusted and tightened up to the required setting, the side play should be tested by a pinch bar, which can be inserted beside the bearing, and if the adjustment is good the connecting rod should be just capable of easy movement along the pin.

Adjustment of Main Bearings.—This is made in the same way as for connecting-rod bearings, and the soft lead wire can be used and clearance measured in the same way for the top brass. The bottom brass can, if necessary, be removed and bedded on the journal or pin, and the clearance on the horns tested. Care should be taken

not to remove metal from the bottom crown of a main bearing brass, because it may thus throw the shaft out of alignment.

Clearance in Bearings.—The thickness of the lead wire taken from the crown of the various bearings should be about—

$\frac{1}{1000}$	inch	per	inch of	diameter of crank pin.
$\frac{1{\cdot}5}{1000}$	,,	,,	,,	crosshead pin.
$\frac{1}{1000}$	,,	,,	,,	main bearing.
$\frac{1{\cdot}2}{1000}$	,,	,,	,,	eccentric sheaves.

The *minimum* clearance for any bearing is about $\frac{1}{100}$th inch, which is about twice the thickness of the film of good oil under pressure.

As a general rule the clearance at the horns should be one and a half times the clearance at the crown, the increase being gradual towards the liners; but the oil-containing strips should have as little clearance as possible. Plummer blocks should also allow some slight clearance at the horns; but the allowance is not so necessary as in connecting-rod bearings, which will not run satisfactorily without a comparatively large amount. When a brass gets warm, the tendency is for it to close at the horns on the pin (as shown in Fig. 434, exaggerated), and consequently nip it closely: which in its turn tends to cause excessive friction, then greater tendency to heating, and next a tendency to close in and nip the pin more tightly. Neglecting to make proper clearance at the horns is a frequent source of over-heated bearings.

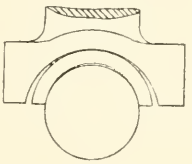


FIG. 434.—Closing of Hot Brasses on the Pin.

Clearance in Piston Packing Rings.—The clearance at the tongue-piece is best determined by calculation of the difference in linear expansion of the circumferential length of the ring and of the cylinder or working barrel. The linear coefficients of expansion of the various metals are given in Chapter III., and the different temperatures of the steam can be obtained from the table at the end of Chapter I. The highest and lowest steam pressures must be allowed for in considering the temperature and linear expansion in any particular cylinder. In addition to the above allowance a standard clearance is allowed, based on experience, of about $\frac{2}{1000}$ths of an inch for each 1 inch of diameter of the cylinder for the same class of material of barrel and ring; that is, say, a cast-iron ring working in a cast-iron barrel or cylinder. Similar clearances must be allowed in cylindrical slide-valve packing rings.

For *piston-rod packing rings* allowance must also be made for wear, and the butts of the segments of metallic packing of the white metal type should have a total clearance of at least $\frac{25}{1000}$ths of an inch for each inch of diameter of the rod. This amount also applies to slide-rod packings.

Thrust Bearings. — The bearing surfaces of the shaft collars should be kept as smooth as possible, and sea-water should be excluded at all times. The clearance between each collar and ring should be carefully gauged, with a properly graduated feeler, to make certain that each ring is taking its fair share of the thrust pressure. Any difference of temperature of the rings when under way points to faulty adjustment. The " ahead " (or after) side of each ring wears away gradually, and thus allows the shaft to move bodily forward ; a mark should be made on some easily accessible part of the shaft, and a gauge made relative to some fixed part of a plummer block or other part, so that the amount of wear can be tested at any time at sea or in harbour. If the wear is excessive, it may be necessary to set the thrust rings aft so as to maintain the axial alignment of the connecting rod central to the mid-length of the crank pins.

Air Pump.—The bucket should be an easy working fit in the barrel, especially when the pump is driven at a rapid rate. The plunger packing should not exert a pressure of more than 1 lb. per square inch of its surface in contact with the barrel, and water grooves should be cut about $\frac{1}{8}$th inch wide at intervals of about 1 inch in the width (axial length) of the packing ring or plunger. The air pump should be kept free from grease, the valves should be frequently cleaned and examined, and the securing arrangements and safety stops carefully inspected before the pump is closed up.

Condenser.—The feed-water filters should be frequently cleaned so as to keep the condenser cooling surface as free as possible from grease. When either soda or potash is used for boiling out the condenser, the mixture of water and soda, or potash, should be carefully excluded from entering the boilers ; the author has tried this method of cleaning, and found it inefficient except when the cooling surfaces can be subsequently scrubbed. The only effective method of cleaning a dirty condenser is by removing the tubes and scrubbing each one separately with a strong alkaline solution.

The protecting plates (mild steel slabs) should be examined, and their electrical connection with the body of the condenser should be renewed at least once in six months. The weed traps require constant

and regular attention, especially in shallow waters. To prevent corrosion in harbour, the air should be excluded as far as possible from both steam and water surfaces of the condenser, and after steaming it should be left full of circulating water, because this is preferable to draining the condenser imperfectly and leaving beads of moisture on the tube surfaces. Corrosion or galvanic action seldom (the author has never met a single case in his experience) occurs on the steam side, and is almost entirely confined to the circulating water side of the tubes.

Setting of Slide Valves, etc.—It is occasionally necessary to check the adjustment of the various bearings in connection with the eccentrics and link motion, and therefore the slide valves may require resetting. The original clearances of valves (and pistons) are usually recorded, and as a rule the original clearances should be maintained, unless there is a very excellent reason for altering them. It should not be necessary to open out either cylinders or slide chests to check these clearances and settings. A gauge should be made, if not already supplied, for each, and marks made to correspond on the rods and guides for certain defined positions of the pistons and valves. (See Chap. XIX.)

Any error in the setting of the various slide valves can generally be detected from the indicator diagram, as detailed in Chapter XIII., with various other defects due to leakage and inefficient working.

Fuel-saving Appliances.—All appliances should be used which can save fuel. Only a certain amount of money can be afforded by the nation, or the company, to which the vessel belongs. If a saving of 10 per cent in fuel can be obtained in ten ships, another one can be kept going. Heat is wasted in many ways—up the funnel by bad stoking ; keeping the condenser too cold, and therefore cooling the feed water below the economical limit of about 110° ; hot drain water is led into the condenser, where its heat is wasted, instead of being used to heat the feed water ; and frequently the pressure in the steam jackets is not properly adjusted to obtain the greatest economy from the steam. It has been already mentioned that the least pressure in any jacket should be equal to the initial pressure in the cylinder ; this pressure can be increased with economy so long as no increase of internal lubrication is necessary.

Bilges.—An engine cannot be kept efficient without cleanliness, although it is possible to have cleanliness without efficiency. The health of every one in the ship depends on the cleanliness of the bilges,

apart from its purely mechanical necessity. There should be no difficulty in keeping the bilges clean when in harbour; and when at sea the parts which are inaccessible should be washed through occasionally, but the water should be allowed neither to rise above the level necessary to effect this purpose nor to spread itself over an unnecessary area. The principal difficulty with bilges arises from carelessness in cleaning out dirt and clearing out the cotton waste used for cleaning, which clog the mud drums and suctions of the bilge pumps, and thus prevent the bilge water being kept below a normal level. The mud drums at the ends of the suction pipes should be *above* the bottom of the floor, so that the suctions are not choked by the dirt drawn or washed into them.

The engineer, before taking over the watch, should, among other necessary duties, inspect the various bilge suctions to see that they are clear. A good routine is for the petty officer in charge of each compartment to report hourly the depth of water in, and state of, the bilges to the engine room. Precaution should be taken, both at sea and in harbour, that all ashes and dirt are cleaned away after the plates have been lifted, so that the platform lies fairly on the supporting angle steels, or floors.

Water Service.—All bearings on which water service has been used should be opened out and cleaned as soon as possible after arrival in harbour. Although the engines may be turned daily, when steam is not up, the bearing surfaces in the presence of moisture soon corrode and become rough.

The water service should be shut off at least twenty minutes before entering harbour, so as to leave only clean oil on the surfaces of the bearings when the engines are finished with. An extra allowance of oil is generally made for this purpose, and, if properly used, saves much work in harbour.

Lubrication. — Breakdowns are generally caused by dirt and corrosion, both of which prevent proper lubrication and contribute to inefficient working of the machinery. Dirt can be kept within moderate limits by constant care, and corrosion is assisted by the presence of dirt. In harbour a lot of unnecessary gritty matter finds its way into bearings and working parts, and is a frequent source of trouble. Care should be taken that such gritty matter (emery cloth, emery powder, and brick dust) is not used near any lubricator or bearing. As a precaution, when getting under way, after a general clean-up has been made and insufficient time is allowed to properly clean all the

bearings and working parts, the water service should be turned on to each bearing to wash out the gritty matter and leave the bearing surfaces free for clean-oil lubrication. The object of all lubrication is to impose a fluid between the two rubbing surfaces, and so prevent them coming into actual metallic contact. The best method of cutting the oil grooves in connecting-rod and other bearings is shown in Fig. 435 (a rough perspective view of the top and bottom brasses) in connection with oil-containing strips.

All lubricators should be scalded out with boiling water and soda, or potash, as late as possible before getting under way, and afterwards at regular intervals when in harbour. The mixture should not be allowed to find its way into the bearings; and two men are required for the work—one to scald through, and the other to attend the lower end of the lubricator pipe. After this cleansing, clean fresh oil should be poured into the cup and allowed to find its way into the bearing. The worsted and the wire to which it is attached should also be cleaned, or renewed, and made of such a length that the worsted will act efficiently as a syphon.

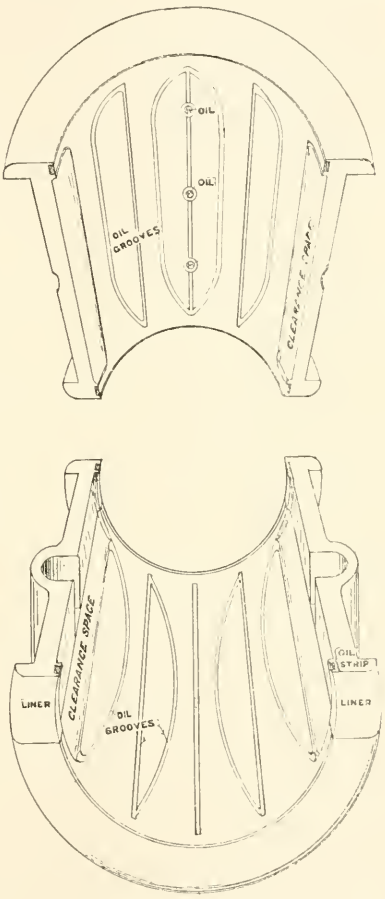


Fig. 435.—Oil Grooves in Bearings.

For external lubrication, the syphon worsted is preferable to any mechanical appliance: it is simple and certain in its action, can be regulated to a nicety, and is very economical. No method, however, compares in either efficiency or economy with a good system of forced lubrication.

A syphon worsted and lubricator box is shown in Fig. 436. The oil cup contains an internal pipe, in connection with an external cock

and short pipe. The oil falls into an open-topped funnel-mouthed pipe, which is led down to the part to be lubricated. This pipe should have a gradual fall, without pockets. The worsted is weighted at the end resting on the bottom of the cup to prevent its being drawn into the internal pipe, and the distance the worsted extends down the pipe is regulated by a twisted wire, which is hooked over the top of the pipe and fastened to the end of the worsted. The end of the worsted inside the pipe should be just below the bottom of the cup. The rate of flow through the worsted can be regulated both by varying the height of oil in the cup, and by varying the number of strands of the worsted.

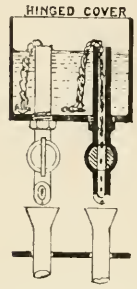


FIG. 436.—Syphon Worsted Lubricator.

Internal Lubrication. — Internal lubrication is unnecessary when condensed or distilled water is used for feeding the boilers in all unjacketed engines, and can be reduced to a very small amount when jackets are in use. Generally no internal lubrication is used for the engines of warships, but a moderate amount of oil finds its way into the cylinders, slide chests, etc., from the working rods and glands. If the air is excluded as much as possible, the working surfaces do not appear to suffer from corrosion in harbour.

Economy by Good Management.—It might be pointed out here that better adjustment, efficient setting of valves and gearing, and good management and training, have enormous influence on the economy of fuel and repairs. Within personal experience, cases have been noticed where increased speed has been obtained, *on less I.H.P.*, in a comparatively old vessel ; and in consequence the " steaming distance " has been increased. In large ships with a well-organised and well-trained staff, accustomed to the vagaries of the special machinery with which they have to deal, it should always be possible to obtain the " specified full power " ; and under favourable conditions of weather and clean bottom, that power should give the maximum number of revolutions per minute originally obtained on the contractor's trials.

A system of changing nearly the whole of the engineer staff in warships, on recommissioning at intervals of two years, is not calculated to obtain good results. This system is productive of a very low mechanical efficiency of the machinery with which a warship is packed, and it appears absurd to perpetuate this old custom of entire change

of crews in ships bristling with mechanisms of every sort and description, and with almost human differences of form and constitution. No organisation or previous experience (other than that of a highly trained engineer) can make up for the lack of training *necessary in the actual ship* for junior officers and men. A better system suggests itself of changing one-third or one-half of the numbers of every rank and rating at annual intervals. What the nation requires of a 25-knot ship is a speed of 25 knots at all times, and this is only to be obtained by a crew accustomed to the actual ship.

It cannot be too clearly impressed on all officers and men that in any successful organisation each individual must play his part, and when left alone, or in charge of any unit or group, must invariably do the right thing, have no hesitation in assuming responsibility, and organise his men and handle the machinery consistently with the application of the great principle of mutual support to other units.

PART XI

MARINE INTERNAL COMBUSTION ENGINES

CHAPTER XXXVII

INTERNAL COMBUSTION, FUELS, AND GENERAL PRINCIPLES

Introductory.—An elementary example of an internal combustion engine is a gun or cannon using some form of explosive as motive power. In this case the combustible is supplied, mechanically mixed with the necessary oxygen for its combustion. Detonation or ignition supplies the necessary local temperature to some small portion of the mixture and starts combustion, which may be either extremely rapid or explosive, or comparatively slow, when it is termed slow burning.

The combustion generates an enormous relative volume of gas from the mixture. In the case of cordite the relative volume is about 700 at atmospheric pressure and at 32° F., and these 700 volumes are compressed almost instantaneously into only one volume behind the projectile. The pressure is still further increased by the rise in temperature, which is about 3750° F. for cordite, of the exploded or burnt gases ; because such increment of temperature produces a corresponding rise in pressure when the volume in which the gases are confined is practically constant (or only slowly increased) in accordance with the law of gases (pressure varies as the absolute temperature when the volume remains unchanged). The total pressure thus sets up motion of the projectile and expels it from the gun, and the energy imparted to the projectile is in exact ratio to the work done and to the energy expended in the bore of the gun. The energy so expended is heat energy, and the gun is simply a form of heat engine which is governed by the same laws as any other heat engine, such as the steam and oil engines and the compressed-air engine in a torpedo.

In internal combustion engines which are used as primary motors for supplying power, a similar action takes place behind a piston in a cylinder. The combustible, and the air which contains the necessary oxygen for completing the combustion, are supplied independently, and the relative quantities are regulated for each explosion ; but as the combustion is comparatively slow, not instantaneous, it is not quite correct to term it an explosion.

Various combustibles are used in internal combustion engines, the principal being town gas (produced from bituminous coal) from the ordinary gas main, producer gas, industrial alcohol, benzol (produced from coal-tar), petrol, paraffin, and other comparatively light oils produced from petroleum, and heavy oils, such as petroleum, almost in a crude state. An oil engine does not differ in principle from a gas engine, but some method is generally adopted either for vaporising the oil into a gas before it is ignited and burnt in the engine cylinder, or for injecting the oil and atomising it under pressure. These methods are explained later, and the general principle on which internal combustion engines work will now be shown.

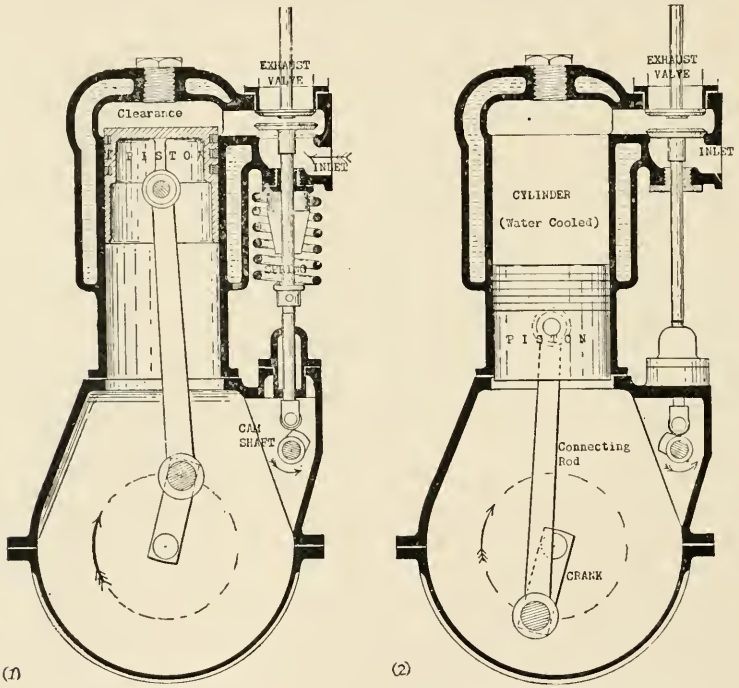
Beau de Rochas, or Four-stroke, Cycle.—This cycle, which refers to a series of operations consecutively performed and continuously repeated while the engine is working, was patented by Beau de Rochas in 1862 ; but no material progress was made until 1876, when the Otto-Crossley engine made its appearance. The cycle consists essentially of a single-acting engine, only one end of the cylinder being used to produce motion, with one working stroke in every four, or one in every two revolutions of the crank. (In the ordinary double-acting steam engine there are two working strokes for every revolution of the crank.) For small internal combustion engines the piston is usually of the trunk type, connected directly with the crank by a connecting rod, and therefore without a piston rod. The four-stroke cycle for a vertical inverted engine (Figs. 437 and 437A) is as follows, in order of the strokes of the piston :

1. *Suction stroke*, during which the air is drawn into the cylinder by the downward movement of the piston through an *inlet* valve. In some engines the combustible gas is drawn into the cylinder at the same time, but in many recent oil engines the charge is usually introduced at some definite instant just previous to the completion of the next, the compression, stroke.

2. *Compression stroke*, during which the air drawn in during the previous stroke is compressed into the clearance space behind the

piston. For gas engines the cylinder clearance space is comparatively large, and the pressure only moderate, from 5 to 40 lb. per square inch ; but for oil engines the clearance space is very small, and in the Diesel engine the pressure rises to about 600 lb. per square inch.

3. *Working stroke*, during which pressure is exerted by the gases on the piston, and work is done in giving motion to the engine. The charge is generally ignited just previously to the end of the com-



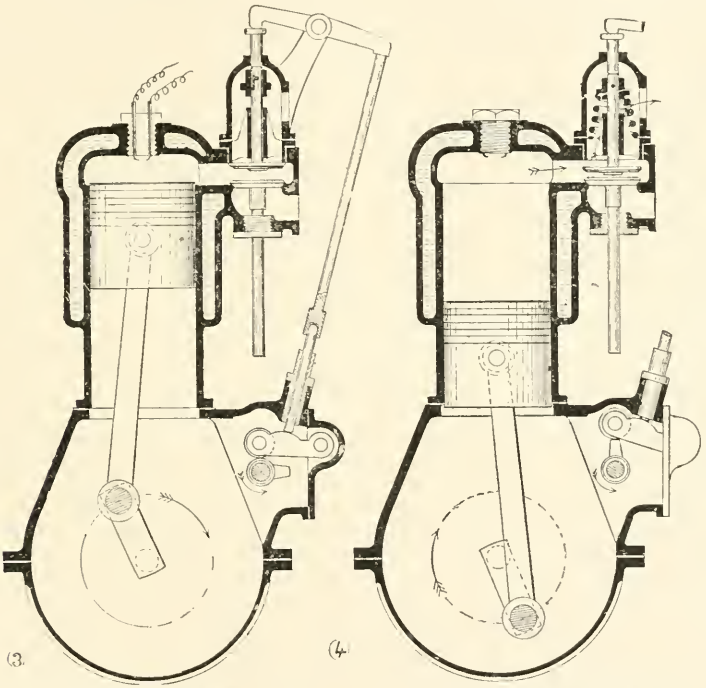
Inlet opens, *suction* begins. Inlet closes, *compression* begins.

Fig. 437.—Four-stroke Cycle.

pression stroke, but not so early as to retard or reverse the direction of motion of the shaft. The combustion produces great heat and temperature, which in turn produce great pressure on the piston which is pushed forward or downward ; during the working stroke the gases expand, and continue to do work as the piston moves downward. At about 0·8 to 0·9 of the working stroke the exhaust valve opens, and the pressure from the cylinder begins to make its escape, generally to the atmosphere.

4. *Scavenging stroke*, during which the products of combustion are expelled from the cylinder by the movement of the piston, but it should be noticed that complete expulsion is never attained, because some products remain to fill the clearance space in rear of, or above the piston. The exhaust valve is closed at about the end of the scavenging stroke, and the inlet air valve opened immediately after.

Since the expiration of the Otto patent in 1890 the four-stroke



Ignition, ready for *working* stroke. Exhaust opens, *scavenging* begins.

FIG. 437A.—Four-stroke Cycle.

cycle has been very generally adopted for shore installations, because the mechanical difficulties of a double-acting engine are not yet entirely surmounted, although considerable progress has been made, especially with two-stroke cycle engines.

An indicator diagram taken from a petrol cylinder working on the four-stroke cycle is shown in Fig. 438. This diagram records the pressure on the back of the piston only ; the crank, or front, end of the piston is open to the atmosphere, and the pressure on it is

atmospheric, or nearly so, and is represented in the diagram by the atmosphere line (14·7 lb. absolute pressure). The suction line naturally falls slightly below the atmospheric line, and the exhaust line naturally rises slightly above the same line. During compression the pressure rises from about 14·7 to 87·5 lb. without ignition, but generally ignition occurs just before the end of the stroke, so that the pressure rises a little, as shown by the diagram near that point. During the working stroke the pressure increased by combustion rises rapidly to about 240 lb. and then falls during expansion, until the exhaust opens and

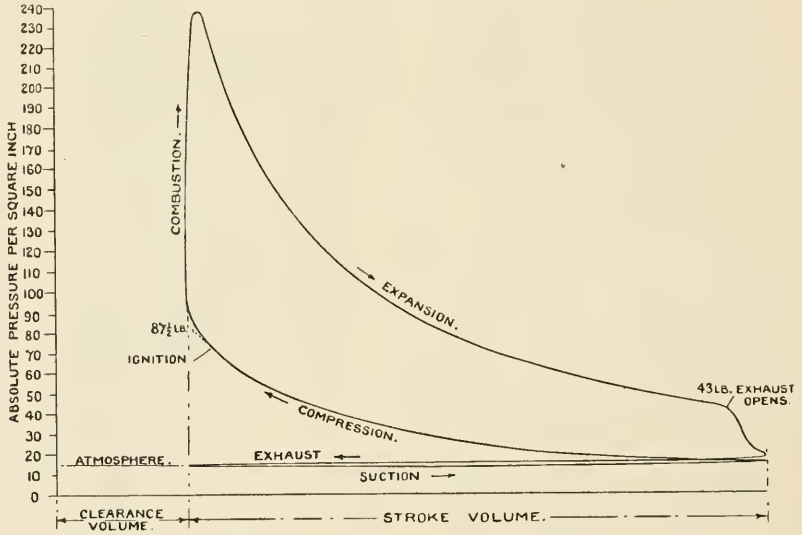


Fig. 438.—Indicator Diagram of Four-stroke Cycle.

allows the gases to escape and the pressure to fall to almost atmospheric. It should be noticed that the pressure attained at the end of the compression stroke (without ignition) is greater when the ratio of clearance-volume to the stroke-volume is less.

Cam or Half-speed Shaft.—In the better types of four-stroke-cycle internal combustion engines the inlet and exhaust valves are mitre seated, and generally spring controlled and operated by the action of the cams on the cam-shaft; some makers still allow the inlet valve to be automatically opened by the suction created by the piston, but as more is now known of the theoretical and practical working of the engine this method is becoming unusual, and rapidly

giving way to the more accurate method of mechanical operation. The position and arrangement of the inlet and exhaust valves should be relatively convenient, the one to the other, for the functions they fulfil.

The cams fitted on the half-speed shaft bear on rollers fitted at the lower ends, in a vertical engine, of the prolonged valve rods. In the Diesel and some other engines the valves are operated by a rocking lever whose opposite end is attached to the valve spindle. The cams, rollers, pins, etc., which form the working surfaces are nearly always case-hardened.

A cam-shaft is generally unnecessary to a two-stroke cycle engine, but in the four-stroke cycle it is revolved at one-half the rate of the crank shaft ; the relative rate of revolution of the two shafts, cam and crank, being governed by toothed wheels or by some form of worm-screw and wheel gearing.

Six-stroke Cycle.—Some actual experiments have been made with an engine working on a six-stroke cycle, in which both the exhaust and suction strokes are duplicated, and thus a purer charge of air is obtained for the compression stroke together with a slightly greater weight of air admitted to the cylinder, because of its lower temperature and consequent greater density. Where weight and space are of no particular importance the slightly greater thermal efficiency obtained by a six-stroke cycle may have some commercial value, but for marine purposes one working stroke in every six of the piston is too small a proportion to compete successfully with the four- or two-stroke cycle, in which, in addition, the engine may be made double acting. Although the six-stroke cycle may show a gain in thermal efficiency, it is evident that the mechanical efficiency must be less than that of an engine working on a fewer cycle and consequently with fewer working parts.

Two-stroke Cycle.—In this arrangement, first proposed and used by Sir Dugald Clerk, every forward, or downward, stroke is a working stroke ; and during every return, or upward, stroke the charge of air or mixture is compressed behind, or above, the piston ready for the working stroke. There is, therefore, one working stroke for *every* revolution of the crank ; in place of one working stroke for every *alternate* revolution, as in the four-stroke cycle.

The action in the two-stroke cycle is as follows :—On completing from 0·8 to 0·9 of the working stroke, the piston uncovers an exhaust port, Fig. 439, and thus allows the products of combustion to escape. A little later in the same stroke the piston uncovers another port which

admits air, slightly compressed, generally to about 5 to 10 lb. above atmospheric pressure ; suitable shaping of a baffle on the end of the piston directs the incoming air, so as to expel, as far as possible, the products of combustion, and to take their place. Shortly after the beginning of the return stroke, the air inlet and exhaust outlet are successively closed by the movement of the piston ; compression now commences, and the cycle of operations begins for a fresh working stroke.

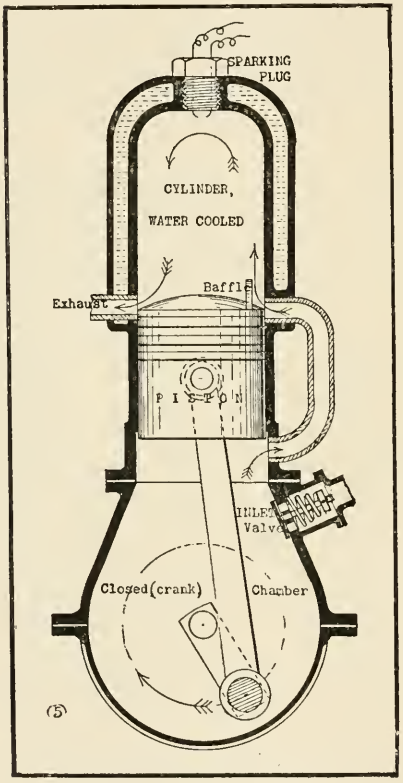


Fig. 439.—Two-stroke Cycle.

Various arrangements are adopted for producing the pressure necessary for scavenging. The most usual is where the crank chamber is enclosed ; on the return, or up, stroke, air is drawn into the crank chamber through a non-return valve, and the down-stroke of the piston compresses this air, and expels some of it into the cylinder through the inlet port, or channel, already mentioned. A higher pressure, under proper direction, renders the scavenging more effective.

In Fig. 440 indicator diagrams are shown, taken from opposite sides of the piston of a two-stroke-cycle engine. The pump diagram, obtained from the front or lower side of the piston, is shown on a larger scale for convenience of illustration. On the back of the piston there is neither a suction nor an exhaust stroke, and consequently the pressure at the beginning of the compression stroke is slightly above atmosphere. The pressure accumulated (without ignition) is therefore slightly higher than in a four-stroke cycle (assuming the proportion of clearance to stroke volume to be exactly the same). The pressure caused by combustion does not usually rise quite so high, because the charge is weaker than in the four-stroke cycle ; this

weakness is due to less efficient scavenging and the presence of a greater proportion of nitrogen.

During expansion the pressure falls according to much the same curve as in the four-stroke, but not to quite the same extent, because the cylinder is slightly warmer and tends to maintain pressure during expansion.

On the pump side of the piston the suction stroke shows less than

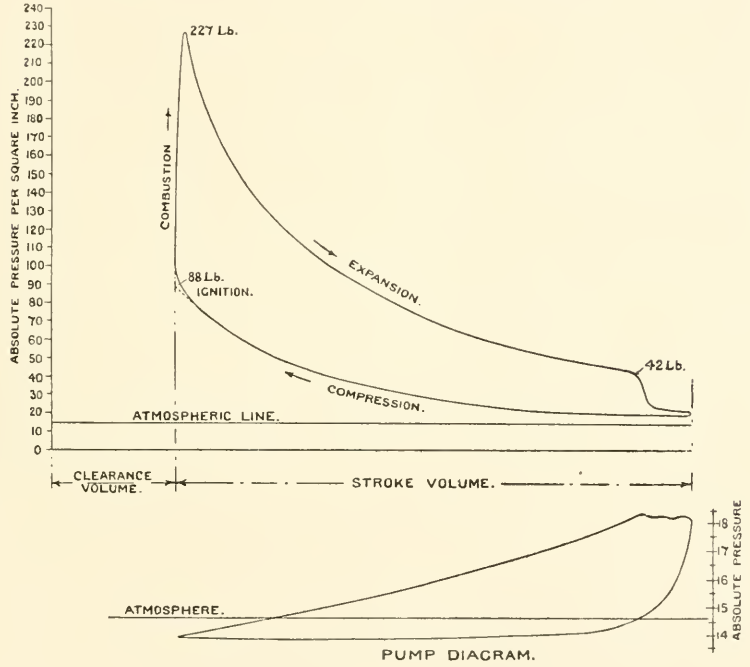


FIG. 440.—Indicator Diagram of Two-stroke Cycle.

atmospheric pressure for most of its length, but at the beginning of the stroke the pressure, above atmospheric, produced by compression during the previous stroke and not utilised in scavenging, falls quickly to atmosphere and below. The pressure accumulates steadily until the scavenging port is opened, when it remains almost constant until the end of the stroke, and then falls rapidly at the beginning of the suction stroke.

In *calculating the mechanical efficiency* of internal combustion engines the area of the pump diagrams of two, four, six (or any number)

stroke-cycles represents a loss. It is usual, however, to deduct this area from the working area when calculating the I.H.P., and to consider the I.H.P. thus obtained as the true working I.H.P.

It might appear from the above description that twice as much power is obtainable from a two-stroke-cycle engine as from a four-stroke engine, having the same-sized cylinder, and under other similar conditions of working. In practice, however, such is not the case, and only about 60 to 80 per cent increase of power is obtained. Some part of the working stroke is inoperative (*i.e.* the part after exhaust begins), and also some negative work is done in compressing the air required for scavenging. The advantage for naval purposes is, however, very great, and even if it is determined to adopt the same *effective* stroke, thus increasing relatively the actual stroke of the engine, the saving in weight and space is very considerable. The thermal efficiency is lower owing to the difficulty of effective scavenging, but considerable improvement has recently been made in this direction.

A double-acting engine, of which there are now many in use, necessarily works on the two-stroke cycle, and the same remarks apply as above. Of course, a satisfactory double-acting engine enables twice the power to be obtained on only a very moderate increase in weight. It involves a piston rod, crosshead and guide, and a consequent increase in height (or of over-all length for a horizontal engine); and difficulties arise in keeping the stuffing-gland tight with the high temperatures used, and in providing some separate means for compressing the air necessary for scavenging. These difficulties are principally mechanical, and therefore may reasonably be expected to be overcome as experience is gained.

Power Gas.—*Illuminating gas* is produced by heating coal in a closed space so that air or other stuff cannot mingle with it, and in this process hydrocarbon gases are given off, of which CH_4 or light carburetted hydrogen or marsh gas, and C_2H_4 or heavy carburetted hydrogen or olefiant gas, are the principal ones. These gases originally formed the only important constituents of ordinary illuminating gas, but recently a limited quantity of both air and steam has been added, thus increasing the volume of gas obtained from each ton of bituminous (gas) coal from 10,000 to 13,000 cubic feet without lowering its illuminating quality, although the heating or calorific value is at the same time decreased from about 550 to 450 b.t.u. per *cubic foot*, at normal atmospheric pressure and temperature, and about 60 per cent is left behind in the retort as carbon or coke. Thus, although illuminating

gas was used extensively as a fuel for gas engines, and is still used where convenience is the first consideration, it is not an economical fuel, and efforts have been directed to still further increase the proportion of producer gas by reducing all the coke to a gaseous form.

For power purposes, therefore, more air is supplied to convert the coke to carbon monoxide, which is combustible (and in the gas engine is converted into carbon dioxide), and the coke is eventually completely burned. Only 4400 b.t.u. are liberated when carbon monoxide (CO) is produced from carbon or coke, and the remainder, 10,100 b.t.u. per 1 lb., can be utilised in the gas engine.

Fuel gases for internal combustion engines are usually produced from solid fuels of a carbonaceous nature, and a gas-producer may be defined as an apparatus for converting solid fuel into combustible gas, which is usually a mixture in varying proportions of carbon monoxide (CO), hydrogen (H), gaseous hydrocarbons (chiefly methane or marsh gas, CH_4), carbon dioxide (CO_2), and nitrogen (N). The first three of these constituents are combustible, but carbon dioxide and nitrogen are non-combustible, and as diluents they lower the flame temperature of the combustible gases when the latter are burnt.

The composition of the gas is largely influenced by the nature of the fuel, and an important consideration is that for all internal combustion engines the gas must be free from tar and other condensable products.

There are two distinct types of power gas-producers : the *pressure* producer and the *suction* producer. The pressure producer works under a *plenum* or excess of atmospheric pressure, which is derived from a fan or pump. The suction producer, which is almost the only modern form, works under suction or partial *vacuum*, and in which generally the supply of air is governed by the demands made on the producer by the engine or other apparatus, such as a metallurgical furnace. Pressure and suction producers differ only in working details, and the gases produced are of similar kinds.

The ordinary stationary gas-producer as used for land purposes is too heavy for marine work, and consumes too much water for scrubbing and feed purposes. It is therefore proposed to describe a special producer which has been invented and developed by Mr. D. J. Smith for motor lorries and other purposes, and which appears to have possibilities for sea service. Full particulars of this development are given in *Engineering*, January 9 and 16, 1920.

There are two principal parts : the producer shown in Figs. 442

and 442A, and the scrubber shown in Fig. 442B. The general arrangement is shown in Fig. 441, and details in Fig. 442C.

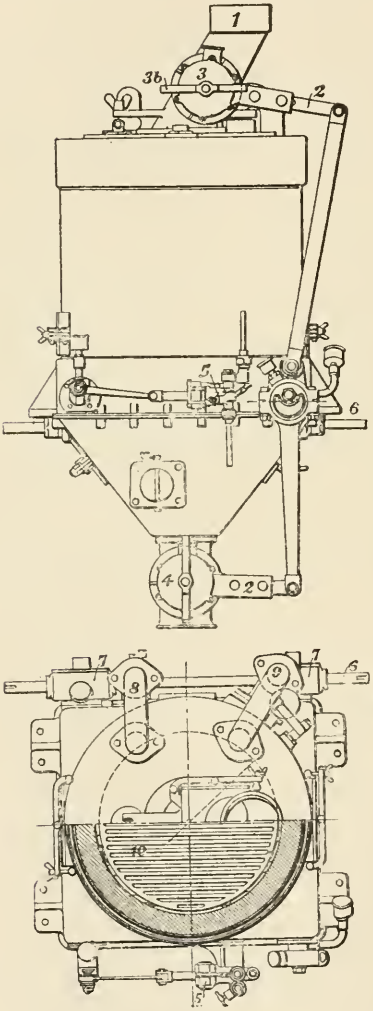


Fig. 442.—Smith Gas-producer.

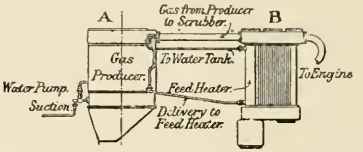


Fig. 441.
Smith Gas-producer: General Arrangement.

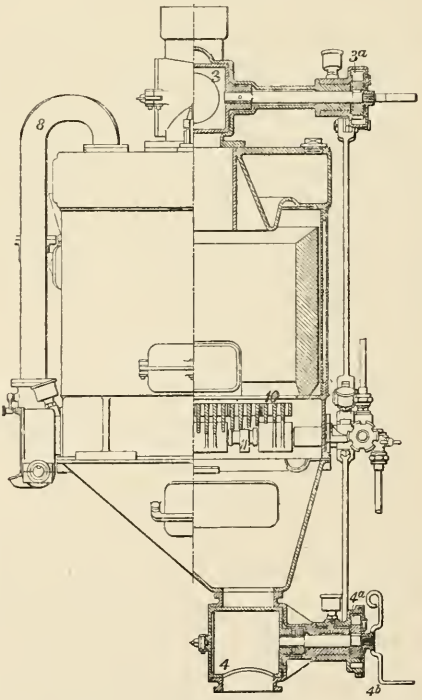


Fig. 442A.
Part Section through Smith Gas-producer.

Gas is produced, by partial combustion and distillation, from some fuel of a carbonaceous nature such as coal, coke, gas oil, maize cobs, or even straw, and in the (dry) scrubber is cleansed of dirt and dust;

it is then passed to the engine, which is of the petrol or similar type but rather larger for equal power, and applied to some useful purpose.

The producer consists of a steel or cast-iron chamber, lined with a special type of light brick-work designed by the inventor, at its lower

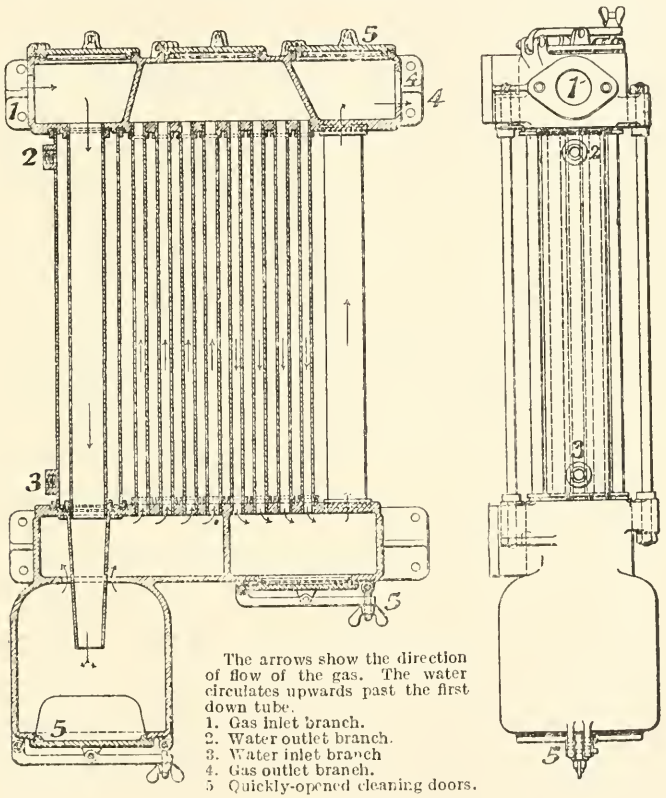

1
2
3
4
5
The arrows show the direction of flow of the gas. The water circulates upwards past the first down tube.
1. Gas inlet branch.
2. Water outlet branch.
3. Water inlet branch
4. Gas outlet branch.
5 Quickly-opened cleaning doors.

FIG. 442B.—Smith Dry Scrubber.

part, and fitted with a fire-grate 10. The fuel is fed into the top of the chamber through the inlet 1, which is fitted with a fuel valve 3, worked in common with the ash discharge valve 4 of similar construction, by the adjustable gear 2. This gear is operated by a friction drive 3*a*, and a handle 3*b* is provided for independent hand operation of the fuel valve. A similar handle 4*b* is provided for the ash discharge.

The upper part of the producer forms a vaporiser or boiler, in

which the normal working temperature is about 200° F., and fed with water by a pump 5 to maintain a constant level. The water from the pump enters the vaporiser by the pipe A (Fig. 442c) so long as the level is below the height of the pipe B ; the excess is led into the cup C and passes down through a pipe to below the fire-bars into a channel in the ashpan filled with asbestos cord which soaks up the surplus water which is vaporised by the heat of the fire-bars. Air enters the vaporiser through the pipe 9 (Fig. 442), and picking up water vapour in its passage, the steam and air are conveyed by a pipe 8 to the under side of the fire, and supply the necessary ingredients for partial combustion. The fire is only 6 inches thick, and there is no clinker. All

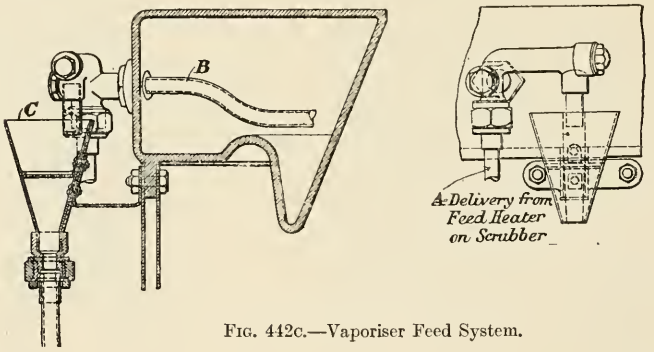


Fig. 442c.—Vaporiser Feed System.

the combustible matter is reduced to producer gas, and a very small proportion of fine ash which drops through the bars into the ashpit below.

The main operating shaft 6 is driven direct from the engine. The driving gear 7 for the cam-shafts is totally enclosed and continuously lubricated. Alternate sections of the fire-bars 10 are pivoted at alternate ends, the free ends being vibrated section by section, and successively by cams on revolving shafts. One of the cams 11 for vibrating the fire-bars is shown in Fig. 442a.

The producer thus briefly described gives off gas free from tarry and condensable constituents of a volatile nature in the fuel, and it is merely necessary to remove the dust held in suspension ; which is easier than scrubbing in a wet scrubber. The Smith scrubber consists of three parts (Fig. 442b), the feed-heater with its inlet at 3 and outlet at 2 ; the cooling tubes through which the producer gases pass upwards through one portion and downwards through the remainder ;

and the filter tubes which precede and follow the cooling process. Each of these filter tubes is fitted with a fine gauze filter, seating in the coned end of the tube. For convenience in cleaning these filters quickly-operated doors 5 are provided, and a sufficient number of tubes fitted so that an arrangement can be made for cleaning one filter at a time without stopping the action of the producer.

The fuel economy is good, with practically no waste ; the weight of the producer plant it is hoped to reduce to possibly no more than 1 lb., required for each B.H.P. of the engine ; and the space is also small compared with a boiler or the older form of producer. Each of these points are of great importance in marine prime movers, and there is a possibility of great development in this direction. As a source of study by engineers it should not be overlooked, because the oil supplies of the world are not inexhaustible, and other sources of power production should not be forgotten.

Industrial Alcohol. — In Germany, France, and the U.S.A., alcohol is produced for industrial purposes from vegetable matter, such as potatoes, and is used as a fuel for lighting, heating, and power. The alcohol is rendered undrinkable by mixing a denaturant (a repulsive ingredient) of a combustible character, such as benzol, crude wood spirit distilled from woods (chemically known as methyl alcohol), and ethylic alcohol.

Industrial alcohol distils over completely between the temperatures of 176° F. and 212° F. Its density before denaturing is about 0·68 to 0·70. It mixes freely with water and produces a solution in any proportion.

Its heat value by weight when denatured or containing 10 per cent of water is about 11,000 b.t.u., and an explosive mixture can be formed with from 8 to 25 volumes of air with alcoholic vapour. When denatured with wood alcohol its boiling-point is lowered from 172° F. to about 158° F., but its latent heat is somewhat higher.

For internal combustion engines a compression of about 180 lb. per square inch can be used without causing premature ignition, which means that a higher thermal efficiency is possible than with petrol and a slightly lower efficiency than with benzol.

Denaturants for Alcohol are (*a*) benzol, in various proportions ; (*b*) German standard denaturiser, $2\frac{1}{2}$ per cent, made up of 4 parts of wood alcohol, 1 part of pyridin, with 50 grammes of oil of lavender or rosemary to each litre ; *or*, $1\frac{1}{4}$ litres of above and 2 litres of benzol with every 100 litres of alcohol ; (*c*) French denaturiser consists in

adding 10 per cent of methylene, which is composed of 75 parts of methylic alcohol, 25 parts of acetine, $1\frac{1}{2}$ parts of heavy benzine, and a small quantity of impurities ; (*d*) acetylene in very small proportion, but this is not recommended because of its tendency to produce acid products, and also it appears to be expensive when made from carbide of calcium. (Professor Spooner.)

Comparison of Petrol and Alcohol, as fuels.—The fuel consumption of alcohol is about 44 per cent greater than petrol per B.H.P., although the thermal efficiency of alcohol is slightly greater, based on an average calorific value of 19,500 b.t.u. for petrol and 11,200 for alcohol. Alcohol is slower-burning, and the mixture with benzol increases the rate of spread of combustion which is necessary for high-speed engines. Although the petrol engine of ordinary design will work satisfactorily with alcohol, it is better to design the engine for use with alcohol if it is desired to use it as fuel. The proportion of the denaturant should also receive consideration. The use of alcohol is dominated by the respective market price, and when it is remembered that alcohol can be produced from any vegetable matter containing sugar and starch, such as potatoes, beet, sugar-cane, Indian corn, rice, wheat, and peat, of which large quantities are available, there can be little doubt of the cheapness of production *if legislation permitted it.* Alcohol is safer than petrol, and its burning can be swamped by water in the ordinary way, which is impossible with petrol.

Benzol.—Commercial benzol is derived from the fractional distillation of coal tar, and is a colourless liquid. When this liquid is heated to a temperature of about 212° F. about 90 per cent distils over, and it is therefore called " 90 per cent " benzol. Distillation begins at about 177° F., which is rather higher than petrol (163° F.). When purified the boiling-point is raised to about 198° F., and its density is increased from about 0·88 to 0·883, or in other words, a gallon of the purified benzol weighs about 8·83 lb. Its freezing-point is below that of fresh water, but by adding petroleum spirit or alcohol its freezing-point can be lowered to about *zero,* Fahrenheit. It mixes freely with alcohol and petroleum spirits and oils, but it will not mix with water.

Its heat value *by volume* is almost as great as that of petroleum spirit ; and, *by weight,* is about 18,000 b.t.u. per 1 lb.

A mixture of 15 per cent with alcohol cheapens the fuel without detracting from its desirable characteristics ; and because alcohol has a future as a fuel for engines (it is much used in Germany) it is probable

that benzol may be selected as a denaturant for preventing fraud on the Revenue in the United Kingdom, when legislation permits the commercial production of alcohol fuel.

For internal combustion engines a compression up to 220 lb. per square inch can be used for benzol without premature ignition, and consequently a slightly higher thermal efficiency is possible than with petroleum spirit, with which a compression of 90 lb. is the usual practical limit. By substituting benzol for petroleum spirit (in the ordinary petrol engine) almost all four-stroke-cycle engines can be worked at half the existing cost of fuel, but for small engines a special carburettor is required. Under working conditions *benzol vapour* requires 36 to 50 times its volume of air for good combustion as compared with petrol vapour, which requires 45 to 80 volumes ; and thus the carburettor must be adjusted to the different proportion to obtain good combustion. Also the clearance space should be less in proportion so as to obtain a greater compression, and therefore a higher temperature, to completely burn the benzol.

As a by-product in the production of illuminating gas, benzol may be comparatively cheap, but it is easy to see that a low price can only obtain in the absence of any moderately great demand.

It is as a denaturant for alcohol that benzol is useful, and beyond this its probably greatly increased price prohibits its extensive use as a fuel for internal combustion engines. The table given below shows a much higher percentage production of benzol from coal tar than given by other authorities, but the proportion per ton of coal is still minute. About 40 per cent of the distillates—that is, all except the pitch residue—is suitable as a fuel for Diesel engines designed for each type, and the late Dr. Diesel was of opinion that even the pitch is capable of efficient use as fuel by adding ignition oil in small proportion.

Dr. E. Muller and P. A. Holiday, in a contribution to *The Motor Boat* of January 25, 1912, state that the fuel consumption with coal-tar oil is slightly higher than when using lighter oils, but the price is lower than that of imported oils. The following table is taken from the same source :—

[TABLE

Order of Distillation of Coal Tar

Fractions of Distillation.	Average Boiling-point.	Average Specific Gravity.	Percentage Weight of Distillation.
Benzol	175° F.	0·9	2
Solvent Naphtha . .	248° F.	0·92	2
Light Oil	338° F.	0·99	2
Karbol Oil . . .	410° F.	1·01	4
Creosote	482° F.	1·04	20
Anthracen Oil . . .	572° F.	1·08	10
Pitch	..	1·3	60
Total . .	..	..	100

Carburation and Combustion of Fuels.—Fuels used in internal combustion engines may be supplied to the engine either in the gaseous or in the liquid state, and in at least one experimental engine solid paraffin fuel has been tried with some success.

Efficient and complete combustion can only be obtained when the air (or other means of supplying oxygen) is thoroughly mixed with the combustibles, and to ensure thorough mixing or *carburation* it is usually necessary to gasify the combustibles if they do not already exist as gases. With liquid fuels there are three distinct processes, viz. : *atomisation*, *vaporisation*, and *carburation*. These three processes frequently appear to be effected in one operation, but this is merely apparent, not actual, and complete combustion without leaving soot or other residue is obtainable only by the co-ordination in proper sequence of these three processes. Combustion in the Diesel type of engine is the best example of this co-ordination, and it is also very clearly evident when using oil fuel burners under boilers.

Bearing the necessity of the above three processes in mind, the next most important influences which tend to produce complete combustion are :

(*a*) *The proportion of air* which is necessary to completely burn the combustible gases. This proportion must necessarily be governed by the chemical composition of the fuel, and it is only within certain limits that the fuel will readily ignite or take fire and burn, as shown by the following table :

[Table

TABLE

Explosion Range of Mixtures of Gas and Air

(From experiments made, by Dr. H. Bunte of Carlsruhe, in a tube $\frac{3}{4}$-inch in diameter; air temperature 32° Fahr., and pressure 29·92 inches mercury.)

Description of Gas.	Percentage of Combustible Gas in the Mixture.			Flame Temperature, Fahrenheit.
	No Explosion.	Explosion Range.	No Explosion.	
Acetylene .	3·2	3·5–52·2	52·4	4262
Alcohol . .	3·9	4·0–13·6	13·7	3362
Benzoline . .	2·3	2·5– 4·8	5·0	..
Carbonic Oxide .	16·4	16·6–74·8	75·1	3812
Coal Gas . .	7·8	8·0–19·0	19·2	3542
Ethylene . .	4·0	4·2–14·5	14·7	3092
Hydrogen . .	9·4	9·5–66·3	66·5	3560
Marsh Gas . .	6·0	6·2–12·7	12·9	3362
Petrol (vapour) .	1·075	1·3– 5·38	5·5	3032

(*b*) *The temperature of the mixture*, whether carburetted as in using *gas* from producers or spirituous fuels, or as a liquid which is merely atomised as in heavy oil engines of the Diesel type, is of great importance to complete combustion, and to the rate or speed with which combustion spreads in the mixture. Assuming the molecular theory of greater vibration of the molecules with higher temperature, then high temperature is an advantage as it promotes both rapid and complete combustion. A high temperature is an absolute necessity, because the hydro-carbon gases of which all fuels contain a more or less large proportion do not dissociate into carbon and hydrogen below a temperature of 1400° F., and therefore cannot be completely burnt into carbon dioxide and steam unless the temperature is greater than 1400° F. Moreover the formation of steam tends to check the rise of temperature, although great heat is given out by the combustion of hydrogen, and it is usually considered that a temperature of at least 3200° F. is necessary to complete the combustion of the fuel.

No advantage is obtained by supplying hot air to a cylinder during the suction stroke, because the weight of air admitted to the cylinder is less with a higher temperature, and consequently the weight of fuel which can be burnt is less per working stroke, and the engine power is proportionately reduced. Also if the cylinder inlet and other

surfaces exposed to the incoming air are hot, they warm the charge which expands and fills the cylinder before a full charge of (cold) air can be drawn in. The inlet opening is therefore proportionately large so that the full charge can be drawn in as quickly as possible.

The outlet or exhaust opening is also large because the retention of the products of combustion after they have completed their work in the cylinder tends to warm the cylinder, which warmth in its turn tends to resist the entrance of the next supply. Apart from the mechanical necessity of keeping the cylinder cool to prevent it from fusing or melting, it is thus within certain limits necessary to keep the temperature low to obtain the full power from the cylinder. On the other hand this removal of heat is a loss, because it is originally supplied by the fuel, and the *thermal* efficiency is consequently lowered in proportion to the heat removed by the cooling water. It is thus necessary to arrive at a compromise, which generally results in regulating the outflow of cooling water so as to obtain a temperature of about 200° F., or just below boiling-point (212° F.).

(*c*) *The compression of the mixture* (or of air alone in some types of engine) promotes high temperature and brings the mixture into closer intimacy. With higher pressure the volume of the mixture is decreased, and consequently the flame has less distance or volume in which to spread, and the cooling surface of the cylinder clearance space becomes less as the compression is increased.

The compression of the charge must not exceed a certain limit which is governed by the nature of the fuel. Thus under ordinary working conditions a carburetted petrol mixture ignites spontaneously at about 90 lb. pressure per square inch, and if such a pressure be attained too early in the compression stroke the engine would be reversed. In many petrol engines the compression would almost reach this limit at the end of the stroke, but ignition usually takes place at $\frac{1}{64}$″ to $\frac{1}{32}$″ before the end of the stroke, when the compression attains about 87 to 88 lb., and at this point, although the pressure eventually rises to 235 lb. from combustion, reversal is not probable although it is quite possible and occasionally occurs. The reversal of the engine may be obtained, and is obtained with certain two-stroke-cycle engines, by igniting the charge much earlier in the compression stroke, so that so great a pressure is obtained within the cylinder that after one or more retarding strokes it overcomes the inertia of the piston and moving parts of the engine, and reverses the direction of motion.

With engines of the Diesel type the air only is compressed, and the timing of ignition is governed by the instant at which the fuel is injected into the cylinder. The temperature of the air is raised by compression alone, and is sufficient to ignite the charge and complete its combustion.

(*d*) *The cooling of the mixture* from contact or close neighbourhood with the close surfaces of the clearance space, cylinder walls, and piston end tends to prevent the spread of combustion, and consequently the surfaces exposed to the charge at the instant of ignition should be as small as possible. In many recent designs the end of the cylinder and piston are concave, so that at the end of the compression stroke the surfaces exposed to the mixture form a more or less spherical shape, which shape presents the least surface area for the greatest volume. Although cooling is an important obstruction to the rapid propagation of combustion, there are many others which affect the efficiency and power of the engine, and particular attention should be given to the design of the clearance space ; and pockets, baffles, and other obstructions should be avoided as much as practicable.

(*e*) *Utilisation of Waste Heat.*—For the highest attainable efficiency it is also evident that the heat required to vaporise the liquid should be obtained from heat otherwise wasted, such as the exhaust waste or jacket waste, and should not be supplied by the fuel itself inside the cylinder, because this supply reduces the working value of the heat supplied to the engine by reducing its temperature and pressure. It should, however, be pointed out that at the time of writing, in the Diesel engine, which attains a very high thermal efficiency, the fuel is injected as a liquid, but heat is supplied by the high degree of compression.

Float Feed Carburettor.—For ordinary petrol engines, and for several paraffin and kerosene engines, a carburettor is generally used. Fig. 443 shows the general principle of a well-known type, which combines in itself the three functions of atomisation, vaporisation, and carburation. The supply of liquid fuel through the pipe A, which is in connection with the fuel tank, is governed by a float connected through levers *CDE*, and a muff coupling *E* with a needle valve *B*. The fuel gravitates into the float chamber, and when the float is raised sufficiently by a change in level of the liquid in the chamber, the float *F* operates the levers *CDE*, fulcrumed at *D*, which push the spindle downwards and close the valve *B*. The supply of fuel to the engine is regulated by the atomiser valve *G*. The vaporising chamber *H*

is surrounded by a jacket space, J, through which a part of the hot exhaust gases are deflected; the proportion of the exhaust gas, and therefore regulation of the temperature, of the vaporiser can usually be effected by a cock fitted near the inlet K. The air enters at L through a wire-gauze screen, and the supply can be controlled as shown

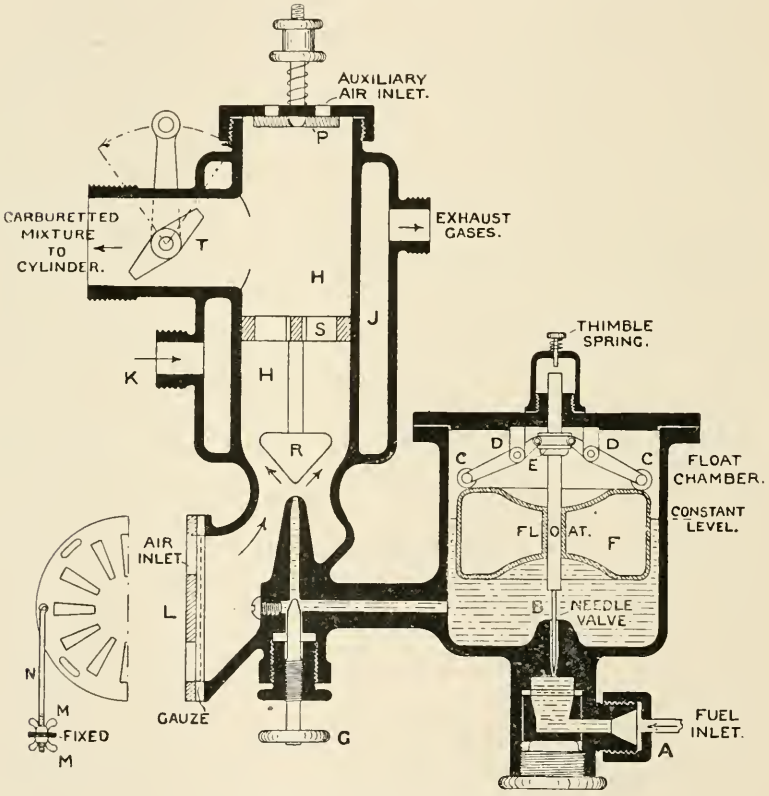


Fig. 443.—Float Feed Carburettor.

by means of two thumb-screws M, on a rod N, in connection with a perforated disc. Additional air can be admitted if required through an auxiliary air inlet valve P, which is controlled by a spring and thumb-screw. The supply of mixture to the engine can be regulated by a throttle valve T, operated either by hand as required or by a governor.

The action is somewhat as follows :—When the inlet valve of the

engine cylinder is opened and the suction stroke begins, a slight decrease of pressure below atmospheric is created in the vaporising chamber *H*. The difference in pressure draws in a small quantity of liquid, through the atomiser valve *G*, which, striking against a roughened cone *R*, is atomised into a fine spray. At the same time air is drawn in through *L*, and mingles intimately with the spray which is vaporised during its passage through the perforated plate *S* by the heat derived from the jacket *J*. Almost complete carburation of the air thus takes place within the upper part of the vaporiser, and a sufficient quantity of the mixture passes from the vaporiser to fill the cylinder. When the cylinder inlet valve closes, carburetted air is left in the upper part of the vaporiser, which forms the nucleus of the next charge for the cylinder, and which, because of its higher temperature, assists in producing more complete vaporisation and carburation. During warm dry weather the hot jacket may not be necessary for carburetting petroleum spirit, but with fuel of higher flash-point, such as paraffin or kerosene, the vaporiser jacket is essential to good carburation when using this general system. In cold frosty weather the vaporiser, if unjacketed, or if not kept warm by a suitable housing, may become choked by ice or frozen moisture drawn into it with the atmospheric air.

Wolseley Float Feed Carburettor.—The Wolseley Company use two main types of carburettors, the two-jet type for pressure and the three-jet type for gravity feed, but the number of jets and the method of supply do not alter the general principles of the system of multi-jet carburettors.

The two-jet type, shown in Fig. 444, has an auxiliary mixture pipe *A* and a small jet *B* for starting and slow running, and a main jet *C* with a separate mixture pipe *D*. For the higher-powered motor cars the mixture pipes are hot water jacketed ; on the small car they are exhaust heated. (In the three-jet type the extra jet comes into operation when the throttle is opened wide for fast running.)

The gradual opening of the throttle valve first opens the small pipe *A*, and a further continued movement opens the second pipe *D* (and in the three-jet type further movement opens the third pipe) until the throttle is full open. There is one float *F* of the buoyancy type to control the liquid supply to all jets.

The extra air valve *G* in connection with the carburettor is provided with a dashpot *K*, and the tension of the spring *L*, holding the valve on its seat, is adjustable by means of a quick thread screw adjustment

M in conjunction with a control lever *N*. This control can be worked from the driver's position. Instructions as to care and working of

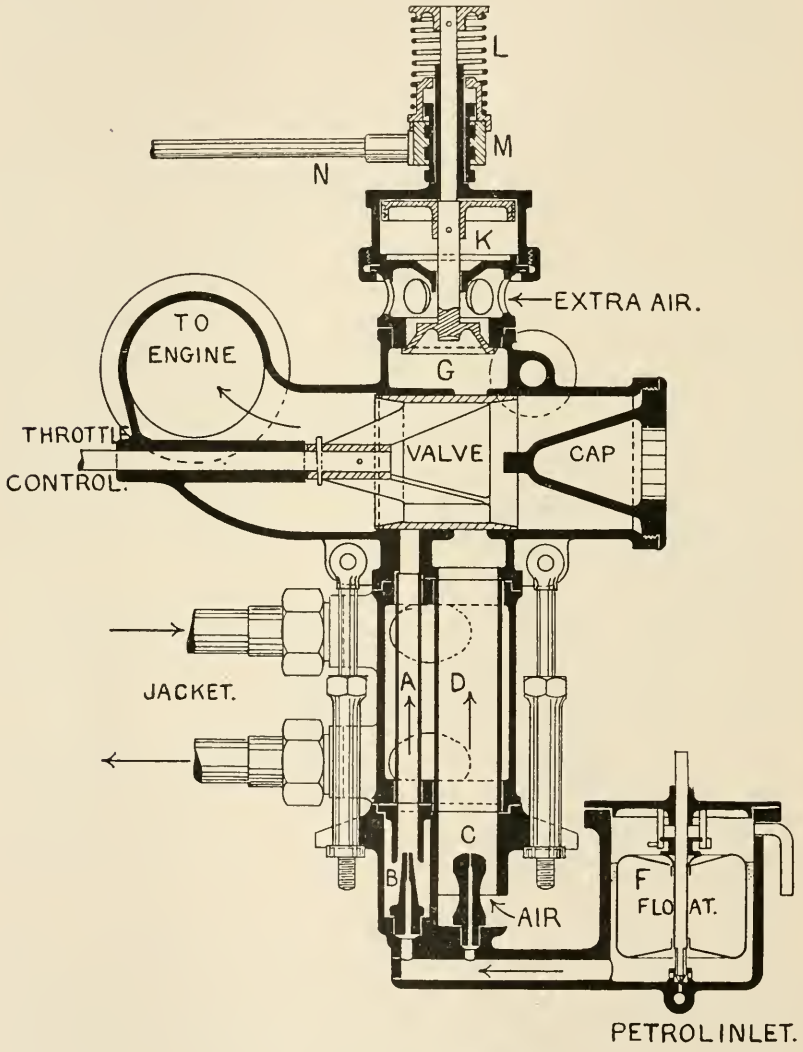


Fig. 444.—Wolseley Float Feed Carburettor.

carburettors are generally supplied by the makers, and it is not considered necessary to describe them in greater detail.

Carburation of High Flash-point Fuels.—In many instances dual

carburettors are fitted side by side ; the engine is started on petrol or other low flash-point fuel, and thoroughly warmed up until sufficient heat is obtained in the jacket of the high flash-point fuel carburettor to permit of its use. The low flash-point carburettor is then shut off, and the high flash-point one brought into use. An alternative method is to heat the high flash-point carburettor by a blow-pipe or other means ; this is not so convenient, but it is safer than using a low flash-point fuel, especially in confined spaces. After the carburettor is thoroughly warmed, no further difficulty is usually experienced with a properly designed engine, but, as pointed out elsewhere, the design of the engine must conform to the kind of fuel required to be used, and no single engine is adapted for efficient use with more than one variety of fuel. The study of the physical properties of the fuel should be the first consideration in the design of the engine, and inattention to this simple rule has caused many apparent failures.

Ignition.—*Ignition* is obtained in four distinct ways :

1. *By Contact with a Hot Flame.*—In the early gas engines the application of a constant external flame to a projecting part of the rear end of the cylinder was made to serve as an igniter or firing arrangement. In its simplest form there was a mechanically operated sliding valve with a single slot cut across it ; when this slot was brought opposite a similarly formed slot in the standing part of the cylinder, the flame of an ordinary gas jet shot through the opening thus formed into the cylinder and fired the charge, which had been previously admitted near the same place. There was always a chance of back-firing with this method, outwards instead of inwards, and consequently a baffle plate was generally fitted to prevent this happening as far as possible. Flame contact firing is now generally discarded.

2. *By Contact with a Hot Tube.*—The tube is generally made of some refractory material, such as metal or porcelain, and in many cases now two tubes are fitted side by side so that one can be renewed while the other is in use. The tube is heated by an external flame which maintains it at a dull red heat (about 900° F.), except at the instant of firing, when it becomes momentarily more or less incandescent. The charge obtains access to the interior of the tube, which is always in open connection with the interior of the cylinder behind the piston. The instant of ignition is governed by a *timing valve* operated by a cam on the cam-shaft ; but for very small engines the admission is automatically controlled through a valve which is incapable of time adjustment, or else there is no valve at all, and ignition is dependent

on the temperature of the tube augmented momentarily by the compression.

In many recent engines the *Hot Bulb* is fitted with an inside plug electrically heated by current from an accumulator or secondary battery. This system allows quick starting, say $1\frac{1}{2}$ to 2 minutes, and is now widely used in engines of the semi-Diesel type for compression of 300 to 450 lb. per square inch; the enormous pressures, 850 to 1000 lb. per square inch, used in the Diesel engine are thus avoided, with their contingent stresses and with practically the same economy on similar fuels.

3. *By Electrical Ignition*, of which there are several varieties. In

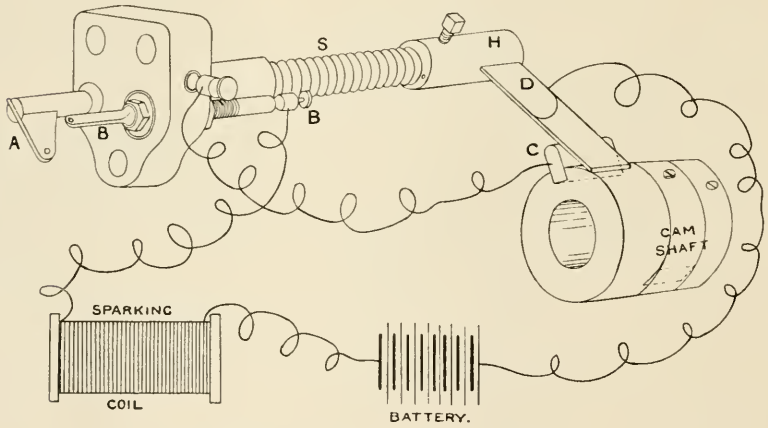


Fig. 445.—Make and Break Electrical Ignition.

the *internal make and break* system, the principle of which is illustrated by Fig. 445, there are two platinum points, *A* and *B*, inside the cylinder head, near the admission inlet. *A* and *B* are wire connected with a cam *C* and a simple sparking coil and battery. *B* is fixed and insulated from the cylinder. *A* is "earthed" and in electrical contact with the cylinder, and, through the rocking spindle *H*, with the tongue *D*. The spring *S* ensures firm contact between the tips *A* and *B* without extreme accuracy of the proportions of the cam, etc., and also gives a sudden separation of the contacts when the cam passes the tip of the tongue *D*. In the position shown no current can pass, but further rotation of the cam joins up *A* and *B*, and a momentary current passes through the circuit, which sets up an inductive current of considerable intensity

(but not a high tension current) in the coil. The sudden release of the tongue and parting of the terminals A and B breaks the circuit, and a fat spark, releasing the current across the terminals A and B, ignites the charge.

In the *jump-spark* system, shown in principle in Fig. 446, an induction coil is used instead of a simple spark coil, and the terminals A and B are both stationary, with their tips from 30 to 50 thousandths of an inch apart; at 400 volts a spark will jump across an air space up to 30 thousandths of an inch. The points A and B are insulated from each other by porcelain, or mica, etc., and are fitted in a metal plug screwed into the head of the cylinder. Wires extend from A and B to the secondary terminals of an induction coil; the circuit in con-

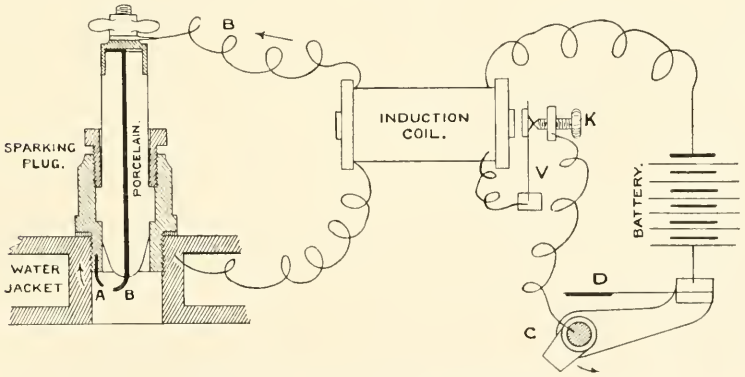


FIG. 446.—Jump-spark Electrical Ignition.

nection with A is usually "earthed." The primary winding receives electric current from a battery (or other electrical source), through a circuit-closing cam C, a tongue D, and a vibrator V. V consists of a steel spring carrying a soft iron piece opposite the end of the soft iron core of the induction coil. When the primary circuit is open at CD, the vibrator spring rests against a contact screw K, which closes the circuit at this point. The primary circuit is completed when the cam C comes in contact with D, and the induction core is then magnetised, and attracting the vibrator V, opens the circuit at K. A rapid vibration occurs and continues so long as C and D are in contact, and the circuit at K is closed and opened by the successive magnetisation and demagnetisation of the induction core. This action produces a high tension current in the secondary coil of sufficient electromotive force

(voltage) to cause a succession of sparks to jump the air gap at the terminals *A* and *B*.

A modification, often used, of the above system dispenses with the vibrator *V*. The secondary circuit being always a closed one, except, of course, for the terminal gap *AB*, contact is made in the same manner as above, a low tension current flowing through the primary winding when *C* and *D* are in contact. At the instant of breaking contact a high tension current is induced in the secondary winding, and a single spark occurs at the gap *AB*.

It will thus be seen that with this system the charge is fired at the instant of separation of *C* and *D*; whilst in the one previously described it is fired immediately after *C* and *D* make contact. Simplification of parts and a greater accuracy of timing are claimed for the single-spark system, whilst with the former system it is claimed that the greater heat produced by the series of sparks will fire weaker charges and ensure a greater certainty of firing.

A diagram of the *Simms-Bosch low tension magneto* system of ignition is shown in Fig. 447. The magnet *M* and armature *N* are fixed, and a soft iron envelope *O* is *oscillated* between them by the movement of a connecting rod *P* worked from a short crank on the end of the cam shaft *R*. When the envelope *O* is in the position shown in the right-hand figure the magnets saturate the armature with magnetism, but no current is induced in the armature winding so long as the envelope stands still. The sudden movement of the envelope *O* into the position shown in the left-hand figure cuts the lines of force between the poles and the armature, which is suddenly demagnetised, and, being instantly remagnetised in the reverse direction, a momentary current is caused in the armature winding of sufficient strength to ignite the mixture. At a certain relative position, generally when the point *S* becomes separated about $\frac{1}{16}$th inch from the point *T*, the current is at its maximum tension, and the two sparking points (of the type shown in Fig. 445) in the cylinder should be separated at the same instant to produce a spark and ignition. By an ingenious arrangement the cam can be moved axially along its shaft on a spiral feather, and the *timing* of ignition can thus be altered within the necessary range.

In the Simms high tension system the armature revolves instead of oscillating, and a sparking plug of the usual type is used, being connected with a secondary winding as in the jump-spark system previously described.

The general principle of a *high tension* (*dynamo*) ignition system is shown in Fig. 448, in conjunction with a Pognon sparking plug. This plug has a space left around the point *B*, so that when ignition takes place a certain amount of expansion is obtained within this space, and it is claimed that the gases, in rushing past the points, tend to clean them. The armature is revolved (not oscillated) either directly by the shaft or through gearing, and is compound wound with both

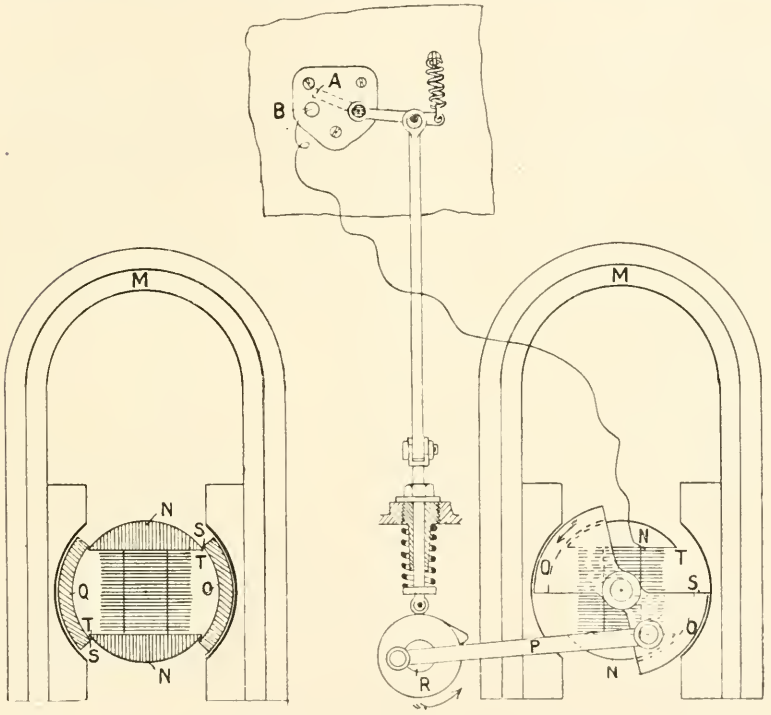


FIG. 447.—Simms-Bosch Low Tension Magneto-Ignition.

primary and secondary coils. The commutator contacts are not shown in the sketch, but can easily be traced when the actual arrangement is seen. The secondary wire is, when working, permanently connected with one of the sparking points, but the primary and thicker wire is connected through a condenser and a contact breaker. The armature and commutator form essential parts of a small dynamo, giving a very high voltage (or tension) with, however, only a very small current strength (ampères), so that the electrical power is actually

very small. The condenser acts as a safety valve to relieve extreme pressure or tension, which may occur under certain working conditions, and thus prevents the possible fusing of the secondary-wire connections. High tension ignition is very reliable, because the discharge jumps across the sparking points easily even when they are dirty, but a very good insulation of the plug is necessary to resist the high voltage and possible leakage. Mica is generally used instead of porcelain for the plug insulation, but several makers still prefer to use porcelain of special quality.

Bosch Dual-Ignition System.—The diagram, Fig. 449, shows the wiring for dual ignition in a four-cylinder (Wolseley) motor, and also the position of a motor self-starting apparatus. The *thin* lines in the diagram show *low* tension and the *thick* lines show *high* tension.

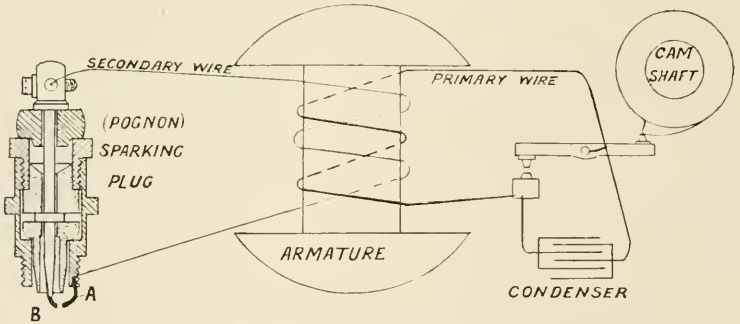


Fig. 448.—High Tension Dynamo-Ignition.

The magneto is of the type already described above, and the armature has two windings—the Primary or Low Tension wire being thick and short, and the Secondary or High Tension wire being comparatively thin and long. Current is also derived from a 4-volt accumulator battery, and is generally used in the first instance before current is obtainable from the magneto, which must be revolved to generate electricity.

With the battery working the positive (or red) current passes through the circuit *EE* to the switch *K*, which can be negatively connected through *F* with the frame and the negative pole *G* of the battery; thus completing the circuit. From *C* the positive current passes through *D* to the distributor *L*, which by a make and break arrangement, worked in positive mechanical conjunction with the armature, operates the spark ignition in each cylinder in succession.

When magneto ignition is used, a current is produced in the primary coil or winding by the rotation of the armature and passes through *BB* to the switch.

The system consists essentially of :

H. 4-volt accumulator battery.

K. Switch for terminal connections.

P. Magneto, or small dynamo.

L. Distributor with connections for each of the sparking plugs.

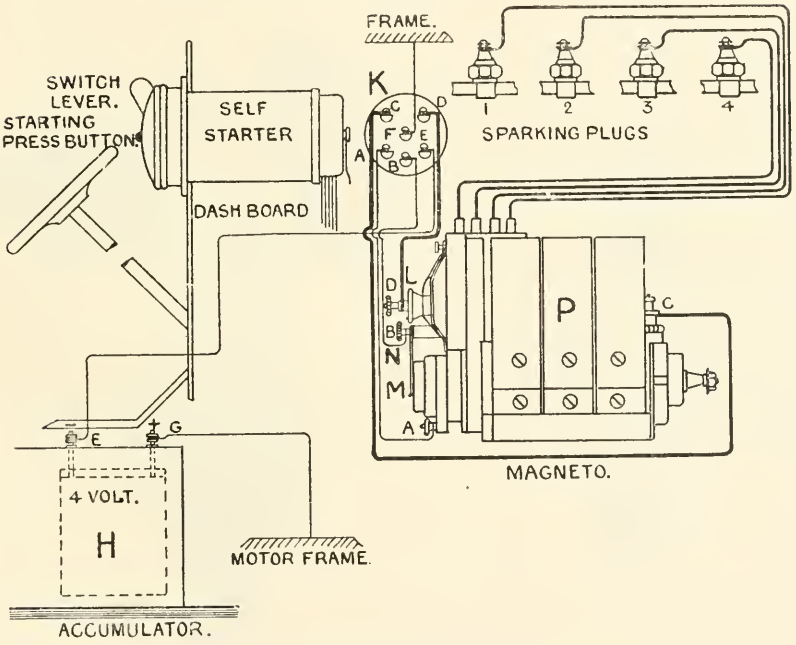


FIG. 449.—Bosch Dual-Ignition System.

M. Contact breaker.

N. Short circuiting connection between *B* and *E*.

With the battery in use, current passes from *H* to *K* through *EE* ; from *K* to *L* through *E* and *D* ; and thence to each sparking plug in succession.

With the magneto in use, current passes from *P* to *K*, through *C* (high tension) ; from *K* to *L* through *C* and *D* ; and thence to each sparking plug in succession.

The distributor is positively driven in conjunction with the arma-

ture, and at defined positions the contact is broken between the terminals *A* and *B* of the primary circuit.

The Lodge System of Ignition depends, in the first instance, on two or more accumulator batteries for power. According to size the 4-volt and 6-volt batteries have a capacity of 20 to 100 ampère-hours, and the current consumption is about 0·75 of an ampère for a 4-cylinder engine up to about 1·25 ampères for a 6-cylinder engine.

The system consists essentially of the sparking plug, a high tension

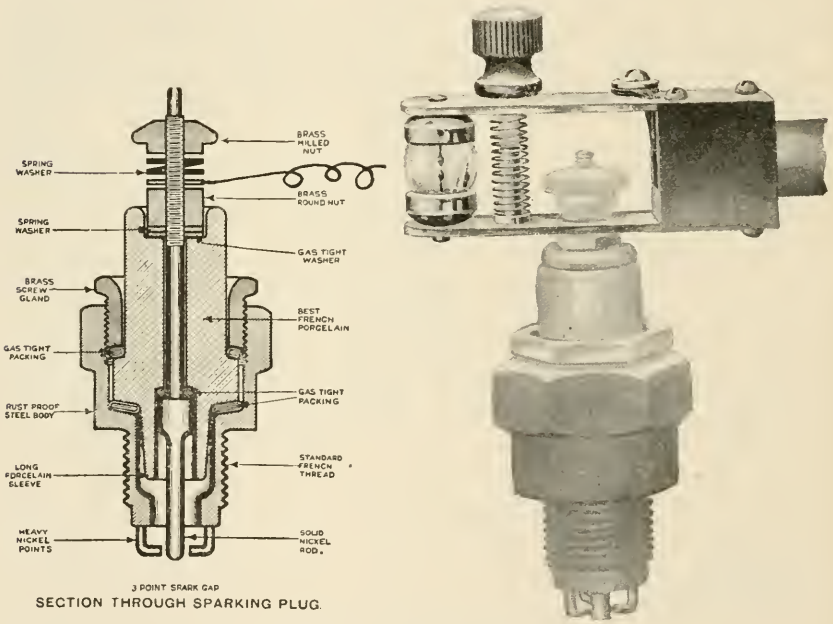


Fig. 450.—Lodge Sparking Plug.

distributor, the igniter, low tension distributor, and the accumulator battery. A Lodge reversing switch is also frequently used.

The Lodge Sparking Plug is shown in Fig. 450, and it can be used equally well with high-tension magneto, coil or Lodge ignition. The sectional sketch almost explains itself, and it will be noticed that there are three points which give three sparks simultaneously. A steel gauge is supplied with each plug for accurately setting the gap between the (electrode) rod and the points.

In the right-hand figure an ingenious arrangement is shown for

testing the ignition; the arrangement is a great convenience, but not essential, and if not fitted, the wire is connected directly with the plug as shown in the left-hand figure. The arrangement consists of a lamp-holder and lamp which are fitted on top of the plug and connected with the H.T. circuit. When the switch is down, as shown in the figure, the lamp is cut out of the circuit; this is the normal working condition. When the switch is raised, the lamp lights up on each occasion of sparking in the cylinder, but if the cylinder does not fire it shows that the sparking plug is short circuited, or that there is no explosive mixture in the cylinder.

The Lodge high tension distributor is constructed for use with petrol engines having two, three, four, or six cylinders, and it enables one trembler coil to fire each cylinder in turn. (It takes the place of the ordinary wire contact shown in Fig. 448.) It is fitted in connection with the cam or half-speed shaft, and consists of a stationary part and casing to which the brushes and plug wires are attached, and of a steel sleeve continuously rotated by the cam shaft. This steel sleeve is divided axially into strips (like a dynamo commutator), and the insulating pieces between the strips are narrower than the brushes, so that the brushes always rest on a smooth steel surface. The correct length of contact is obtained by the setting of the brushes.

The Lodge igniter is made in various designs, in which, however, the general principle is the same, and is shown in Fig. 451. The secondary winding terminals of the sparking coil are connected with the interior of two Leyden jars or high tension condensers. The outer coatings of the jars are short circuited by a leak or imperfect conductor, the object of which is to keep them always at the same potential except at the instant of a sudden electric rush. The leak relieves the leads and sparking plug of excessive stress, and the terminals remain at the same potential up to the last moment, when the two jars are full and overflow at *A*. At this instant everything is liberated, and with a rush of inconceivable rapidity the jars empty themselves across *A*, and round the complete circuit through the sparking plug. The rush is

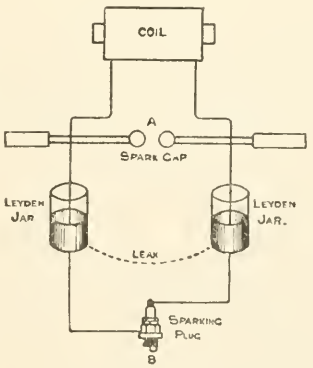


FIG. 451.—Lodge Igniter.

over in a millionth part of a second, but not before it has ignited the combustible mixture exposed to the sparking plug.

The rush is so violent that not only is dirt in the path blown away, but the electric momentum overshoots the mark, and the jars are charged up in the reverse direction by the impetus ; they then discharge again, and again are charged in the ordinary way, and so on, many times, without the coil taking further part in the action ; its function is over when it has filled the jar to overflowing. The *A* spark is a pioneer spark which precipitates the sudden rush and causes the *B* spark at the sparking plug.

In the general construction of the Lodge igniter, the *A* spark gap is under glass and always in view, as is also the trembler which is of essential importance.

The Lodge low tension distributor is of the wipe contact type, and

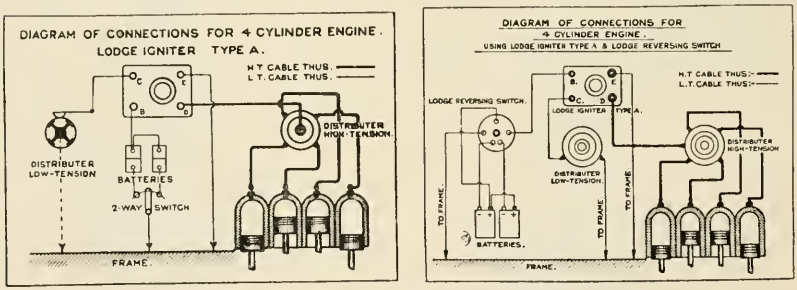


Fig. 452.—Lodge Diagrams of Connections.

built on the same lines as the high tension distributor. It is used to govern the timing of the ignition.

The Lodge reversing switch is of no use with magnetos, but with accumulator batteries it is used to automatically reverse the direction of the electric current, and thus saves pitting and wear of the platinum points.

Diagrams of connections for a four-cylinder engine are shown in Fig. 452, with and without a reversing switch. The diagrams would be similar for six cylinders except that a 6-form, instead of 4-form, distributor is used.

Ignition by Compression.—The compression of air or gas tends to increase its temperature. If the compression is done very quickly, so that no heat can escape by radiation, conduction, etc., from the containing surfaces, the curve of compression would conform to the " adiabatic " curve, *CE* in Fig. 133. If the compression is done very

slowly, so that all the heat generated by compression escapes from the air or gas, and no change of temperature takes place (the compression being then isothermal), the curve is of the hyperbolic order, *CD* in Fig. 133. In actual practice the compression is rapid, but some heat escapes, and therefore the curve of pressure falls between *CE* and *CD*, and with a fast-running engine is generally more nearly in accord with *CE*. The average compression curve conforms fairly closely to the equation

$$P \cdot V^{1\cdot3} = \text{constant}.$$

Engines using ignition by compression are usually of the *Diesel* type, in which under normal working conditions compression of air to 500 lb. per square inch produces a temperature of 947° F., depending greatly on the rate of motion, the temperature of the water jacket, and the rate of conduction from the interior of the cylinder to the external water (in the jacket).

The degree of compression depends on the nature of the fuel: for petrol 90 to 100 lb. would be sufficient, and probably this would very nearly suffice for alcohol, benzol, and some producer gases, but for oil of high flash-point, generally 200° F. and above, much higher compressions are possible, and higher temperatures can be adopted with advantage.

The comparative degree of compression is decided on previous practice in conjunction with the ratio of the stroke volume to the clearance volume of the working cylinder. The clearance volume behind the piston in Diesel engines is usually about 7 per cent of the volume struck out by the piston in each stroke, and thus during compression 107 volumes of air are compressed into seven volumes, producing about 500 to 600 lb. pressure per square inch, and a corresponding temperature of 1000° to 1450° F., according to the working conditions.

In the Diesel engine the fuel is sprayed into the cylinder at the end of the compression stroke, and therefore the charge (of air) cannot ignite prematurely. Combustion takes an appreciable time, and instead of a peak of high pressure as shown in the diagrams, Figs. 438 and 440, for a petrol engine, the highest pressure continues for an appreciable part of the working stroke (Fig. 464). In an ordinary petrol engine a temperature of 2750° F. may obtain for an instant after ignition, and in consequence the exhaust temperature is high (about 1250° F.); with the Diesel engine the maximum temperature seldom exceeds 1500° F., and the exhaust temperature is comparatively low

(600° F. about). For a low temperature it is easy to design an engine, but for high temperatures, approaching the melting point of the material, it becomes difficult, especially with cylinders of large diameter. A description of the Diesel engine is given later.

The *instant of ignition* is timed to occur a little before maximum compression, which without ignition would occur at the end of the compression stroke. The exact instant in practice depends on the speed of the engine, and is best determined graphically.

In some engines the point of ignition is stated to be about $\frac{1}{64}$ to $\frac{1}{32}$ of an inch before completion of the compression stroke, but some difficulty is experienced in determining the position of the crank on such a small dimension, and it is better to proceed by determining the position of P, the point of ignition on the crank circle, Fig. 453.

Combustion takes an appreciable time, depending on the conditions of pressure and temperature of the mixture, the proper chemical proportions of the mixture, and the method and intensity of ignition. For a petrol engine under good working conditions $\frac{1}{200}$th part of a second is about the average time taken for completing combustion, and it is usual to consider that this period consists of two equal parts, one before and one following the completion of the compression stroke, during which the crank (see Fig. 453) will travel from P to N, where $PT = TN$. As a matter of fact the latter part of the combustion is much more rapid than the early part. If r equals the length of the crank CP in feet, n equals the number of revolutions *per second*, then the crank centre moves at an average velocity of

$$2 \,.\, \pi \,.\, r \,.\, n \text{ feet per second,}$$

and therefore combustion will continue while the crank centre moves through

$$\frac{2 \,.\, \pi \,.\, r \,.\, n}{200} \text{ feet, from which}$$

$$PT = \frac{\pi \,.\, r \,.\, n}{200}, \text{ in which } \pi = 3{\cdot}1416 = 3 \text{ approximately.}$$

$$\text{If } n = \frac{1000}{60} \text{ revolutions per second (a common speed),}$$

$$\text{then } PT = \frac{\pi \,.\, r \times 1000}{200 \times 60} = \frac{\pi \,.\, r}{12}$$

$$= \frac{r}{4} \text{ approximately.}$$

It is evident that the distance PT increases in direct proportion to the rate of revolution, and thus for high speeds the instant of ignition

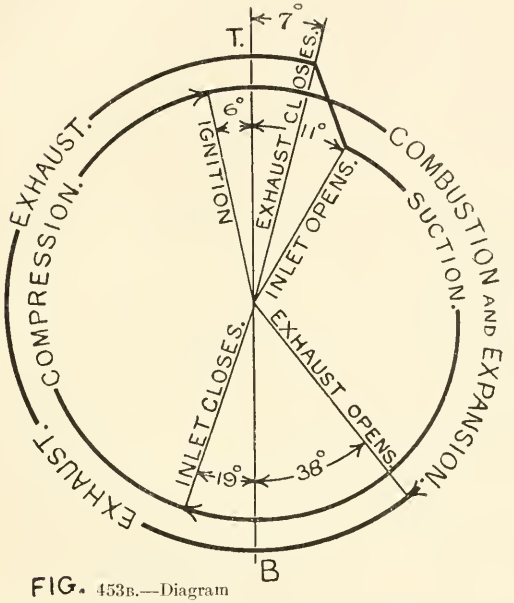


FIG. 453B.—Diagram showing Relative Positions of Operation in 4-Stroke Cycle Engine.

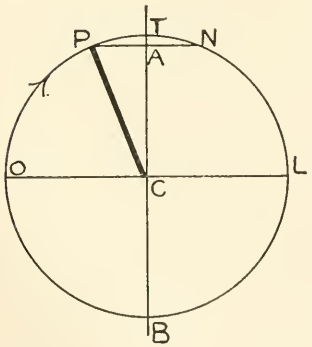


FIG. 453.
Instant of Ignition.

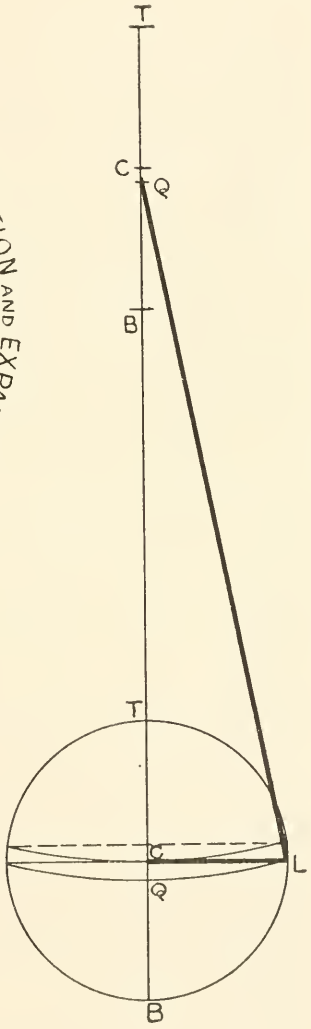


FIG. 453A.
Relative Positions of Piston and Crank.

should be earlier and for low speeds later than for the normal speed. In many instances a timing apparatus is fitted for this particular purpose.

With a length of connecting rod equal to five times the length of the crank, the angle $PCT = 4°$, when $AT = \frac{1}{64}''$
$= 6°$, when $AT = \frac{1}{32}''$
$= 7\frac{1}{2}°$, when $AT = \frac{1}{16}''$.

Referring to Fig. 453A, if the crank is in the position CL, the piston will not be relatively at C but slightly below C; and if the length of the connecting rod is five times the length of the crank, the piston would be 0·0505 times the length of the crank below the mid-stroke position, which is relatively the point Q. This difference is technically known as the error due to obliquity of the connecting rod, and is greatest at about mid-stroke; near the ends of the stroke the error becomes less and is generally negligible.

For naval (steam reciprocating) engines, the length of the connecting rod is almost invariably equal to *four* times the throw (or length between centres) of the crank; for mercantile machinery it is somewhat greater, from $4\frac{1}{4}$ to $4\frac{3}{4}$; for motor engines, five cranks appear to be the general practice, but with long-stroke engines probably the naval practice of four cranks will be generally adopted. The lengths of stroke and connecting rod determine the height or length of the engine, and to an appreciable extent greater height or length means greater weight. Within certain working limits it is therefore important to keep the length of stroke and length of connecting rod low, but below these limits there is a considerable loss of efficiency and power.

In Fig. 453B, a diagram shows the relative positions of the various operations in a four-stroke-cycle engine of common construction.

Cooling.—It is necessary in all internal combustion engines to cool the cylinder, the cylinder end, and the valve casings which are generally attached to it, for two reasons: (*a*) to prevent overheating of the metal which, when the temperature is increased above about 600° F., is rapidly reduced in its strength; (*b*) to allow efficient lubrication, though with oils now obtainable having a flash-point of 550° to 600° F. it is not so difficult of attainment as in former times. For the same reasons it is necessary, in large engines running at high speeds, to cool the piston and rod. The upper limit of temperature allowable for the metal of the internal working parts is thus about 550° F., and even this comparatively moderate limit is seldom reached.[1]

It is also necessary to make the piston and its packing rings fit fairly accurately in the working barrel, to prevent leakage. With the

[1] Cooling by internal injection is referred to later.

high temperatures obtained from the explosive gases, the piston, being unable to radiate its heat so easily as the cylinder, expands in a greater degree, and therefore becomes too tight a fit in the cylinder to obtain a high mechanical efficiency. The objection to cooling is that heat is abstracted from the cylinder, and, in its turn, from the heat energy contained in the gases produced by combustion; consequently the lower the limit of temperature to which the cylinder is cooled the greater is the loss of heat. The "heat rejected" by cooling amounts to about 30 to 45 per cent for cylinders of 20 H.P., and for larger cylinders is less, possibly only 20 per cent of the total heat energy contained in the fuel. Lower cylinder temperature has the advantage of higher *power* but with *less* economy of *fuel*.

For motors of three to eight horse-power, such as used for cycles, and for aviation, *air cooling* is generally adopted. The cylinder body (and end, in many cases) is made with a number of thin, deep ribs or rings, projecting, generally radially, from the body and from the end. These ribs being made in one piece with either the cylinder or its end, rapidly radiate heat away to the surrounding atmosphere. With motor cycles, rapidity of motion through the air increases the cooling action; and in cases where the motion is insufficiently rapid for this purpose, a fan, driven by the motor, is used either to impel air on to or to draw it past the cylinder. An example of insufficiency of air cooling is when a motor cycle is climbing a hill and its maximum power is required; great heat is given out by the engine, which the lack of draught from the slow motion of the cycle fails to convey away, and the cylinder warms to a temperature approaching a dull red heat of about 900° F. Air cooling has also been adopted for motors of much larger horse-power, and for these a large fan power may be necessary; some considerable saving in weight is made in this way over the water cooling next described. In cold climates air cooling is preferable to water cooling, because the trouble of draining the water jackets to prevent fracture by freezing when the motor is not in use is avoided. The saving in weight is, however, generally effected at the cost of efficiency, because a considerable fraction of the power developed by the engine is absorbed in driving the fan.

Water cooling is generally used for engines of over eight horse-power. The cylinder body or end, or both, are made with jacket spaces through which water is circulated. The cooling outfit, in addition to the jacket spaces, generally consists of three parts: a tank, a radiator, and a pump; and, occasionally, a filter or oil separator.

The pump is usually fitted in the coolest part of the system of circulation, which is just before the water enters the jacket spaces ; it should be of a type either centrifugal or rotary, or preferably helical, which does not materially retard the flow of circulation should it fail for any reason.

For locomotive work, where the quantity of water is necessarily limited, the same water is used over and over again by cooling it during circulation through a radiator and tank fitted in series with the cylinder jacket. Radiators vary in design : a simple type, still to be seen in many motor cars, consists of a number of gilled tubes (or of one continuous gilled tube folded backwards and forwards over itself) placed in the most exposed position, so as to meet with the greatest possible air draught when the car is in motion. The hot water from the jackets passes into the radiator, where it gives up its heat to the metal of the tubes, which in turn radiate it to the atmosphere. A fan is often fitted in conjunction with the radiator, partly to assist the draught when the car is moving, but principally to cause one when the car is at rest with the engine running. With stationary engines, if the supply of water be limited and a continuous stream is not available, the tank is made large enough to act as its own heat radiator, and neither gilled pipes nor fans are necessary.

For marine work the tank is usually unnecessary, and a continuous supply of cooling water can be obtained from outside the vessel ; but for continuous sea-work special arrangements must be made for removing the lime scale deposited in the jacket when sea-water is used and high jacket temperatures are maintained. A circulating pump is frequently unnecessary, and in well-arranged water-cooling systems natural circulation is often relied on. Perhaps the best example of natural circulation is afforded by that of a properly designed kitchen boiler hot-water plant with its pipe and tank connections.

With a vertical inverted cylinder the natural direction of circulation is upwards from the lower end of the cylinder body jacket into the end jacket, and then to the upper part of the radiator, where the cooling assists the downward circulation to the tank and pump. This direction of circulation is generally adopted for the Diesel engine (see Fig. 455), and is a suitable one because the ignition or firing is dependent on the temperature produced by compression alone ; the hotter water is near the point of ignition. Any excessive cooling action (which might possibly be produced if the direction of circulation were reversed)

tends to reduce the temperature obtained by compression and so prevent proper ignition. The colder circulating water, surrounding the lower end of the cylinder, tends to reduce the temperature, and therefore the pressure during the working stroke. The adverse effect produced by this cooling action is, however, in engines where the charge is ignited by compression, counteracted by the comparative slowness of the combustion which, by what is known as " after-burning," tends to maintain, and does in practice generally maintain, the pressure in the cylinder above that obtained by a more nearly instantaneous combustion, notwithstanding the comparatively more rapid cooling towards the end of the working stroke.

In other engines, looking at the problem from a general point of view, where an independent form of ignition, such as electrical, is used, it appears preferable to adopt a circulation in the opposite direction, *i.e.* first through the end jacket, and then through the body jacket from back to front (or top to bottom). In this way a lower maximum pressure and a greater mean pressure may be obtained, because the cooling effect of the water jacket may cause slower combustion after firing, while the higher temperature of the jacket water near the end of the working stroke has less effect in reducing the temperature during the stroke. With a vertical cylinder the reversal of the natural direction of flow means that a pump must be employed, but with a horizontal cylinder natural direction of circulation can be followed.

The tank can be placed as convenient, but as it is important that the water jackets and the radiator should be completely filled, the level of the water in the tank should be above the highest points which require to be filled. For this purpose a vertical pipe with an open mouth is frequently fitted to the tank, and serves also as a filling pipe ; air-cocks or plugs, as convenient, are fitted at the highest points in the circuit to allow any air to escape when filling. In some instances a tank is non-essential, and its place and function, which is simply to automatically prevent overheating and to make up loss by evaporation, is taken by the vertical filling pipe, the open mouth of which prevents an accumulation of pressure.

The usual temperature of the water on leaving the jackets is about 170° to 180° F., but in some instances higher temperatures are allowed and some steam is formed ; a pressure above atmospheric is seldom permitted, although there is a growing tendency to utilise the water jacket as a boiler to supply steam under pressure for atomising and

vaporising the oil on entering the cylinder. It is claimed for this system that an increase of efficiency can be obtained, because heat, otherwise rejected to the cooling water, is returned to the engine and does work on the piston. Without the results of practical experiment, any forecast as to the possible increase or decrease of efficiency due to this method is entirely speculative. Assuming the mean temperature of the jacket water to be about 150° F., and the mean temperature of the products of combustion near the working barrel to be about 550° F., the mean temperature of the cylinder metal is probably about 350° F., which is a temperature which could be increased without danger to the metal and with some increase of working pressure during expansion.

The effect of steam or *water injection* during the combustion is

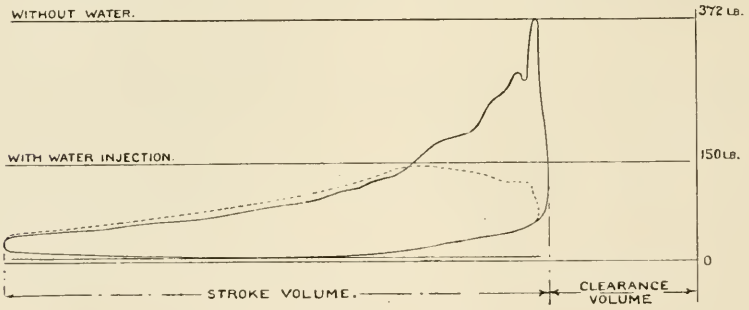


FIG. 454.—Effect of Water Injection.

shown by the indicator diagram in Fig. 454. The full lines show the ordinary diagram obtained without steam or water, and the dotted lines the effect of introducing a slight quantity of water, which is almost instantly converted into superheated steam of about 150 lb. pressure. Instead of a high peak of pressure, there is a tendency towards a more uniform pressure throughout the working stroke ; the temperature in the cylinder is probably less, and, if too great a quantity of water or steam is introduced, the combustion may therefore be less complete ; there is about the same mean pressure, but on account of the lowering of the maximum pressure the cylinder need not be so strong, and a considerable saving in weight might be effected if the system could be relied upon ; and in consequence of the more uniform pressure during the working stroke the turning moment, or torque, on the shaft is more uniform. This system also commends itself as suitable for a double-acting engine, because the lower pressure and temperature

reduce the mechanical troubles of gland-packing, lubrication, and piston and rod cooling.

Cooling by Internal Injection of Water.—In a paper read before the Institution of Mechanical Engineers by Professor Bertram Hopkinson, F.R.S., at Cambridge, July 29, 1913, particulars are given of a successful experiment on internal cooling by an injection of water into the cylinder of a 50 B.H.P. Crossley gas engine of the horizontal type. The usual water jacket is entirely omitted, and the engine was fitted with a new cylinder for the test. Reference is made above to a system of water injection *with the charge*, and such water is principally evaporated at the expense of the *charge* and the heat generated by its combustion. The Hopkinson method aims at direct cooling of the heated metal of the cylinder, piston, and valves exposed to the heat of combustion. The injection water, therefore, derives its heat for evaporation from waste heat, and not from the charge which is almost unaffected by the injection. In the system described above there may be some gain which is principally mechanical, but in the Hopkinson system there is more than the mechanical gain because there is almost no thermodynamic loss, and any gain from the formation of steam is devoted to increase in work done. Expenditure of water for internal cooling is not economical in marine practice, because some method must be devised for condensing it after use, and returning it to the engine ; this objection is, however, not insuperable, and if the system offers advantages in other directions, such as a reduction in cylinder stress, better lubrication indirectly, and possible increased diameter of cylinder within the same safe working limits, the condensing apparatus might follow as a natural sequence. Where water is plentiful or obtainable cheaply in sufficient quantity, the Hopkinson system appears to have many advantages, and generally no pump would be required.

Compression Stroke.—Both during the admission and after the inlet valve is closed, heat passes into the charge from the cylinder walls, etc. ; the pressure rises, and therefore the work done during compression is increased. This work of compression is done by the fly-wheel, or other external agency, at the expense of energy previously accumulated in the mechanism, and therefore increases the wasteful expenditure of energy during compression. Evidently, if the original temperature is high, the final temperature becomes very much higher ; and the liability to premature ignition, which frequently occurs for this reason, is increased. On the other hand, after the temperature

is raised, by compression, above that of the cylinder walls, heat is given out to the walls (energy is thus wasted), but there is a tendency to decrease the temperature otherwise due to compression in the later part of the stroke. Ignition is generally made to occur before the end of the compression stroke, but the exact position depends on the nature of the fuel and the speed of the piston ; the fuel, it might be again stated, is usually already in a gaseous state at the instant of ignition. Ignition should be so timed that the maximum temperature is attained before the piston has completed one-twentieth of its working stroke, but it should not be early enough to exert any appreciable retarding effect on the turning motion of the shaft. (In certain engines, such as the Ailsa Craig type, where the crank axis is not in the same plane as the cylinder axis, the ignition is usually arranged to occur later than in the ordinary type.)

Working Stroke.—As the piston begins to move forward, combustion continues and tends to maintain the highest pressure attained by constantly adding heat ; but the movement of the piston rapidly increases the volume, and consequently the pressure falls after about one-tenth of the stroke is completed. In many cases the fall of pressure begins with the stroke, and is due to the increase in volume not being compensated by increase in temperature. It should not be forgotten that at the very high temperatures which are easily produced in internal combustion engines, carbonic acid gas, or carbonic anhydride (CO_2), and water vapour or steam (H_2O) become dissociated respectively into their chemical elements, and heat is taken from the gases to effect these dissociations ; temperature and pressure therefore tend to fall for this reason. In some later position, when the temperature has dropped considerably, these elements, very possibly, again become chemically united by combustion ; exactly the same quantity of heat is then returned to the gases as was previously extracted to effect their dissociation. The above is probably the reason for the apparent addition of heat during the working stroke, which is a matter of common observation during experiments, though it may also be due to " after-burning " and slow combustion.

After-burning is more or less common with all types of gas and oil engines, and is a continuation of combustion after the first instant of ignition or firing. It tends to maintain the temperature and pressure, so that the actual expansion curve, as shown by the indicator diagram, is more nearly adiabatic, and in some cases above the theoretically adiabatic curve. If after-burning is too prolonged, or if the stroke

is too short, some portion of the fuel passes away unconsumed with the exhaust products, and thus occasions a loss : it is generally caused by too great an admixture of air, the mixture then becoming too weak in fuel.

Scavenging Stroke and Exhaust Control.—The exhaust valve, if one is fitted (it is unnecessary in the ordinary two-stroke-cycle engine), is nearly always operated mechanically by a cam on the lay shaft. The instants of opening and closing are often capable of adjustment while the engine is working.

With exhaust control, as with ignition and admission of the charge, the element of time is of first importance to efficiency. There are now many useful data which show the time necessary for the several operations. Only a general statement can here be made, but recent experiments should be studied in relation to the speeds at which the crank and piston are moving respectively.

The exhaust should be opened as late as possible to avoid the too early escape of the exhaust products, which may still have some energy left that has not been used up in doing work on the piston ; in certain cases a retardation of the exhaust opening has resulted in a gain of efficiency and improvement in power. At the dead points the piston moves relatively only a small distance while the crank travels through a comparatively large angle ; it is near this point, therefore, that the necessary time required for exhausting can be best given without losing either power, which might possibly be otherwise developed in the working stroke, or compression, which is only very slightly diminished.

Engines with a short stroke and great speed of revolution are liable to great loss by exhaust waste ; that is to say, by the escape of the exhaust before all possible work has been obtained from it. With high piston speeds, say about 700 feet per minute, the exhaust valve is generally opened at a point between 8 and 9 tenths of the working stroke and closed just after the beginning of the suction stroke.

Back-firing.—Pre-ignition caused by incorrect timing of the ignition apparatus produces a too early explosion in the cylinder which is technically known as back-firing or firing back. This results in a tendency to reverse the direction of motion of the shaft by preventing the completion of the compression, or upward, stroke. Even if reversal is not effected on the first occasion of firing back, it is likely to occur during the next compression stroke or strokes. In many engines working on the two-stroke cycle, pre-ignition obtained by alteration of the timing is used to effect reversal of direction of motion

of the shaft. In other engines, although no alteration in timing is made, reversal is caused automatically after the engine is thoroughly warmed and the load is removed or considerably decreased. Back-firing occurs in the cylinder, but other troubles are caused by external firing which are enumerated below.

External Firing.—Troubles of this kind are usually attributable to—

1. *Short circuiting* of the sparking apparatus, which produces ignition at the wrong instant.

2. *Bad ignition* from weak electric current, dirty sparking plugs, or imperfect mixture or proportion of the combustible and air. In both cases a charge of gas and air is expelled into the exhaust pipe without being properly fired, and this is ignited when the exhaust valve is opened by the incandescent products of the next charge which is ignited. Ignition and firing occurring in the exhaust passage and pipe is technically termed *exhaust explosion*, and is particularly liable to occur after a miss-fire.

3. *Derangement of the valves*, when hot-tube ignition is used and the charge is fired before the inlet valve is closed ; but if the valve is already closed and ignition occurs too early, the trouble is an internal one and is actually back-firing, as pointed out above.

4. *Leaky inlet valve* allowing the charge in the cylinder to communicate with the charge outside the cylinder. Firing a charge in the mixing chamber is sometimes attributable to slow combustion and a flame lingering in the cylinder until the inlet valve again opens. In a very bad case the flame may reach the petrol or gas-pipe conveying the charge to the carburettor, and for this reason gauze discs are usually inserted across the induction pipe to minimise the risk.

5. *Over-lubrication* may produce, when comparatively low flash-point oils are used for the purpose, a too rich mixture from vaporisation of the lubricant. Flame lingers in the cylinder and, when the inlet is opened, fires the charge prematurely. Over-lubrication, however, is usually the cause of considerable waste, and in many cases the unpleasant smell of the exhaust is attributable to it as well as to imperfect combustion of the charge. Also gritty carbon particles are left on walls of the cylinder and piston which lead to excessive wear and friction, and in a very bad case may cause the piston to stick in the cylinder.

Cylinder Lubrication.—Steam turbine engines require no internal lubrication, and reciprocating steam engines of the condensing

type require no more than finds its way into the cylinders, etc., through the rod glands and packing. The internal combustion engine, is, however, primarily dependent for successful working on the efficient lubrication of the piston packing and cylinder, and failure in this respect is still extremely prevalent, although steady improvement is being made. Few troubles are now experienced in small engine practice, but the larger oil engines appear to suffer more by comparison. Internal working conditions do not differ very greatly as regards temperature of explosion, or the maximum temperature at which combustion takes place, which ranges from 2900° to about 3200° F., and it is evident that this temperature in itself does not account for failure. The degree of rapidity, or rate, of combustion increases with the pressure generated, and is greatest in those engines which use fuel of the highest flash-point, but opposed to this natural phenomenon is the generally larger diameter of cylinder associated with high flash-point oil engines as compared with the low flash-point petrol and spirit engine, and consequently the temperature reaching the walls of the working barrel should differ little in the various classes of engine, although large *gas* engines appear to have some advantage from utilising both a low flash-point fuel and large cylinder diameter.

The difficulty in keeping the piston cool is intimately connected with the construction of the cylinder and piston. The compression end of the piston is regularly exposed to the highest temperature of combustion, but at similar regular intervals the other end (in a single-acting engine generally) is allowed to overrun the cylinder, and becomes exposed to the cooling influence of atmospheric air and splash lubrication. This cooling effect is of great importance, although it tends to reduce thermal efficiency slightly by conduction through the metal of the piston and radiation from the internal surface, but this loss is balanced by decreased friction between the piston rings and cylinder barrel. Oil engine practice has hitherto involved cylinders of small diameter compared with steam practice, in which, however, the smallest piston is generally relieved of the friction consequent on the obliquity of the connecting rod by transferring it to an external guide (Fig. 141).

In steam practice a pure mineral oil for cylinder lubrication is necessary, because generally vegetable and animal oils contain some acid or acid-producing qualities which eventually attack the boiler and other internal surfaces with which they come in contact. These conditions are not applicable to internal combustion engines in the

same degree, and the very slightly acid nature of vegetable and animal oils has insufficient time to do any material damage to the working surfaces of the cylinder interior and piston rings. A pure mineral oil is naturally a fuel, and, under circumstances favourable to its combustion, it burns away and leaves no film of lubricant on the working surfaces. Vegetable oils are also combustible, but in a less degree than mineral oils, and consequently castor oil, rapeseed oils, and olive oils are frequently used either in a nearly pure state or compounded with mineral oils as a cylinder lubricant. Pure castor oil appears to be the most favoured for air-cooled engines of the Gnome type, and a heavy viscous mineral oil is almost invariably used for petrol engines as fitted in motor cars and lorries. Between these extremes there are many compounds of vegetable and mineral oils which find a ready market as cylinder lubricants. It is generally considered that it is advisable to extract a large proportion of any paraffin wax which the original crude mineral oil may contain before it is used as a lubricant.

CHAPTER XXXVIII

MARINE INTERNAL COMBUSTION ENGINES

THERE are a few examples of gas-producer engines used for marine purposes, but these may be considered to be entirely experimental and not at present likely to compete seriously with either light or heavy oil engines.

Another development is the Still engine, which combines internal combustion with a steam engine without separate boiler plant. The Still engine has considerable possibilities and will be considered at the end of this chapter.

At present to be considered there are three principal types of marine internal combustion engines :

(1) **Petrol or Gasolene Engine.**—This is a larger and more powerful type of the motor car engine, and several of the fast launches built during the war have been fitted with engines of the airplane type because others were not immediately available. The Coastal Motor Boats (C.M.B.'s) were the most powerful and speedy and were fitted with engines of about 375 H.P., making a speed of over 40 knots.

The motor launches, or M.L.'s, of which about 600 were built and in service during the war, were fitted with two sets of six-cylinder engines developing a total of about 220 H.P., or less than 20 H.P. per cylinder. Although all these vessels have been used at sea for war purposes, they cannot be considered as actual sea-going vessels and are not likely to be used in time of peace for trading and other purposes, and therefore hardly come within the scope of this book.

For boats carried in larger vessels the motor boat has practically superseded the old and cumbersome steam boat, but the motor is now frequently of the paraffin type, which is started on petrol and run on paraffin after warming up, and is accordingly fitted with a dual arrangement of carburettors. From what has been said in the previous chapter there should be no difficulty in understanding the working of

these small motor engines, of 5 to 40 H.P., and the various parts can be seen and understood.

(2) **Paraffin Engine.**—Engines of rather larger power than stated above are frequently fitted in fishing vessels to work on paraffin or heavy oil of about 0·82 density, and are in many instances of the semi-Diesel type using hot-bulb ignition. These engines are generally non-reversible but have an arrangement, of the motor-car type, of a lever attached to spur gearing, for reversing the direction of rotation of the propeller. For small powers there is no particular objection to reversing gearing but it is obviously unsuited to large powers. These engines will be understood from what has been stated in the last chapter and any further information is covered by reference to what follows on the oil engine of the Diesel type.

(3) **Heavy Oil Engine.**—In these engines, *pure air* is compressed in the cylinder, and the oil is introduced just before the turn of the stroke. Ignition is produced entirely by temperature obtained by compression of the air, and thus the compression must be greater than that required to attain this temperature. In practice allowance must be made for a compression of about 500 to 600 lb. per square inch, and after ignition takes place the increase of temperature may increase the pressure in the cylinder to 1000 lb. per square inch, which for purposes of design of the cylinder is generally assumed as the *maximum* working pressure, but in a well-proportioned engine the actual maximum pressure in the cylinder seldom exceeds 600 lb. per square inch.

Power per Cylinder.—The power per cylinder of all internal combustion engines is limited by considerations of the stresses occasioned by the maximum working pressure in the cylinders of possibly 1000 lb. per square inch in heavy oil engines, and ranging down to 250-300 lb. in petrol and gasolene engines, combined with high temperatures of about 3000° F. in all light and heavy oil engines.

In large-powered engines the cast iron cylinder is fitted with a *steel* liner, thus relieving the cast iron to a great extent from the special stresses arising from working conditions. The design of this liner is of great importance in its relation to the power to be obtained from each cylinder, and a few remarks will not be out of place here.

For present materials the ultimate tensile strength of the steel material may be taken as 27 tons per square inch, but progress and experience may allow an increase to 32 tons or more, and such an increase is desirable to effect a decrease in the number of cylinders necessary.

The proof stress or load at which the material yields and may obtain a permanent set is about one-third of the tensile strength, or for a tensile of 27 tons is about 9 tons. Allowing for a suddenly applied load, only about one-half the proof strength should be taken as the allowable maximum working stress, or (call this stress, T):

$$T = \tfrac{1}{6} \text{ of } 27 \text{ tons} = 10{,}000 \text{ lb. about.}$$

If d = diameter of the liner in inches,
t = thickness of liner in inches,
p = water pressure test $\times 2 = 2000$ lb. per square inch,

then, referring to Fig. 28, Internal Pressure, for illustration of principle,

$$2 \,.\, T \,.\, t = p \,.\, d \;.\;.\;.$$

from which
$$d = \frac{2 \,.\, T \,.\, t}{p} = \frac{2 \times 10{,}000}{2000} \times t$$

and
$$d = 10 \,.\, t.$$

And thus, when t is 1″ . . . d becomes 10″
2″ . . . ,, 20″
3″ . . . ,, 30″
4″ . . . ,, 40″

And so on for material whose ultimate tensile strength is 27 tons.

The practical and experimental consideration is: What thickness may be safely used, when taking into account the critical temperatures and conditions imposed by them to cause rupture from unequal expansion of the inner and outer surfaces of the liner subjected to the greatest changes? So far as present experience goes a maximum thickness of 3 inches is the safe limit, and thus the cylinder for a material of tensile of 27 tons must not exceed 30 inches in diameter. If we assume a higher tensile of, say, 32 tons then we obtain a cylinder of safe limit of $35\frac{1}{2}$ inches diameter.

From results of recent practice, and proportionately because cylinders of these sizes have yet to be tried over extended periods of working and are still considered experimental:

About 385 B.H.P. is obtainable from a cylinder 30 inches in diameter.
,, 500 ,, ,, $35\frac{1}{2}$,,
,, 1000 ,, ,, 50 ,,
,, 2000 ,, ,, 70 ,,

And the respective thicknesses of liner for 27 tons tensile would be

about 3″, $3\frac{1}{2}$″, 5″, and 7″, and for 32 tons tensile would be $2\frac{1}{2}$″, 3″, $4\frac{1}{4}$″, and 6″.

These great thicknesses are only required in the region of the greatest pressure, for say one-tenth of the stroke in normal practice, as shown in the illustrations later, and the liner may be of only a moderate thickness for the remainder of its length. Also, as will be seen from the illustrations, the liner may be reinforced by a great thickness of cast-iron cylinder near the region of maximum pressure, but this exists principally for its proper purpose of supporting the cylinder head or cover and for transference of the working stress to the engine framing. With very large cylinders difficulties may arise from contingent causes such as ignition, scavenging, and the necessary valve fittings in the cylinder head; observing that large valve areas would be necessary and the requirements would have to be met by a number of small valves of moderate dimensions to prevent leakage from distortion.

In the formula $$d = \frac{2 \cdot T \cdot t}{p}$$

with improved materials and greater experience in actual practice it may become possible and with a sufficient margin or factor of safety to increase T from 27 tons to say 36 tons and to reduce p to 1500 lb. and then

$$d = \frac{160}{9}\, t = 18t \text{ nearly.}$$

This would admit of the cylinder liner being constructed over 50 inches diameter, under 3 inches in thickness and developing upwards of 1000 B.H.P.; in this direction future progress is practicable for obtaining greater power per cylinder, and a reduction in the number of cylinders required for great powers in fast vessels.

In this estimate of power per cylinder it is necessary to consider other factors which have a great effect in its determination. Compared with a double acting steam engine, which has two working strokes to every revolution, there is only one working stroke in every alternate revolution of a four-cycle internal combustion engine and the formula for the I.H.P. becomes

$$\text{I.H.P.} = \frac{P \cdot L \cdot A \cdot N}{2 \times 33{,}000}.$$

With a ratio of stroke to diameter in internal combustion engines

of 1 to 1, which is common in high speed engines of the submarine type where head room is of great consequence, the indicated mean pressure during the working stroke may be taken as about 108 lb. per square inch. If the ratio of stroke to diameter is 3 to 2, which accords with mercantile practice, the indicated mean pressure is reduced to about 85 lb. per square inch. Again, in the internal combustion engine the B.H.P. is usually considered, and the mechanical efficiency being considerably less than in the steam engine, an efficiency ratio of 85 per cent is seldom exceeded, so that the actual Brake effective pressure in the above example becomes about 68 lb. per square inch.

The piston speed, L, N in the above formula, is also a factor, which in mercantile practice is much the same as in reciprocating steam practice and may be put at about a maximum of 960 feet per minute, while in submarine practice it is considerably greater and of the order of about 1250 to 1400 feet per minute.

All the above factors, including thickness of metal dependent on tensile of material and the factors of safety employed in design, the ratio of stroke to diameter which affects the mean pressure in each working stroke, and the piston speed, affect any result which may be calculated. A few examples from recent practice may be useful to point the moral of these remarks :—

S.S. "Glenapp."—Cylinder diameter 29·92″, stroke 43·31″, revolutions per minute 125, piston speed 902 feet per minute, and B.H.P. 330 per cylinder.

German Submarines of high speed and short stroke to give head room. Cylinder diameter 20·86″, stroke 20·86″, revolutions per minute 380, piston speed 1310 feet per minute, and B.H.P. 292 per cylinder.

Both the above are four-stroke-cycle engines. Examples of two-stroke-cycle engines are :—

Sulzer type, experimental : Cylinder diameter 39·4″, stroke 43·31″, revolutions per minute 150, piston speed 1125 feet per minute, and B.H.P. 2000 per cylinder.

The comparative success of the above engine resulted apparently in the evolution of a standard type engine for land purposes :

Sulzer standard type land engine : Cylinder diameter 29·5″, stroke 39·4″, revolutions per minute 130, piston speed 855 feet per minute, and B.H.P. 600 per cylinder.

Cammellaird-Fullager opposed Piston Type : Cylinder diameter 18·5″, stroke 25″, revolutions per minute 130, piston speed 458 feet per minute, and B.H.P. 500 per cylinder. (Two pistons being

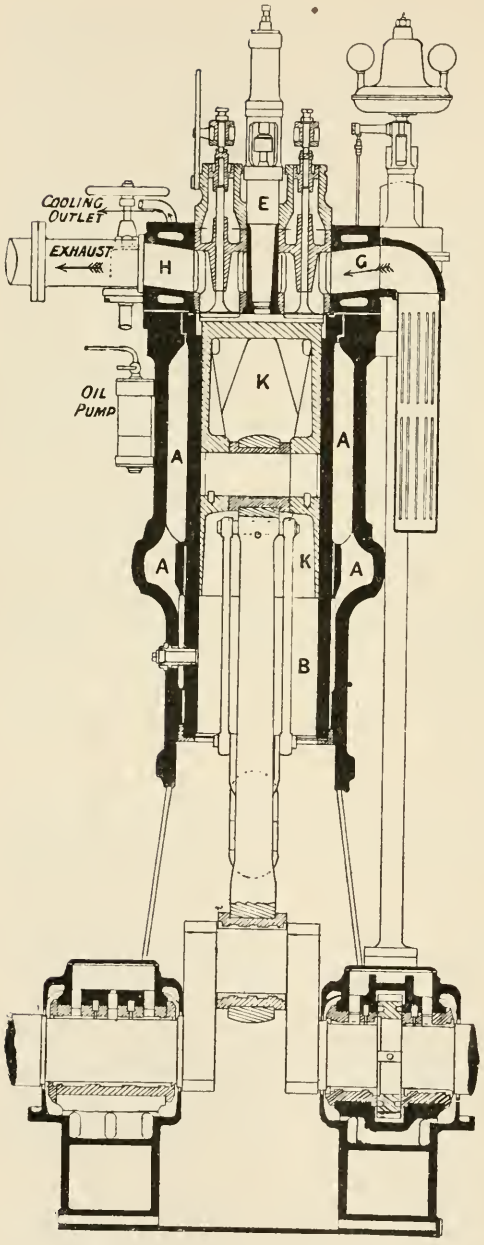


FIG. 455.—Diesel Four-stroke Engine.

fitted in each cylinder, the equivalent effective piston speed is about 916 feet per minute.)

Diesel Engine.—The engine, Figs. 455 and 456, works on the four-stroke cycle and is usually vertical. The upper part of the frame forms a water jacket *A* which encloses a cylinder *B* of steel or close-grained cast iron. The cylinder cover is deep and hollow, and the space *C* forms a jacket into which the cooling water enters through *D* from the cylinder jacket space. Near the middle of the cover is fitted the oil sprayer *E* (for detail see Fig. 462), and near it is a starting valve *F* through which compressed air from a storage reservoir can be passed. On the right is the air inlet *G*, and on the left the exhaust *H*. All the valves are spring-loaded. Both the air inlet and the exhaust outlet are kept closed by pressure in the cylinder, and are opened by bent rocking levers actuated by the cam shaft *T*. The air-compressing pump *L* is water-jacketed; just before the end of the engine's

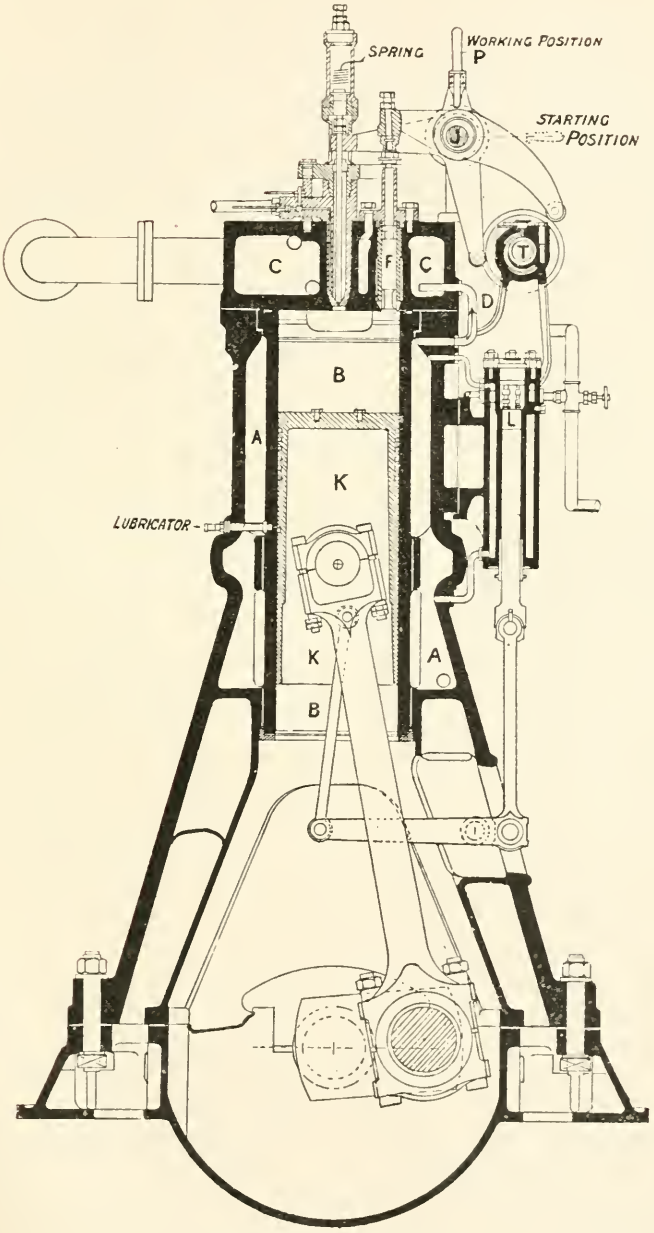


FIG. 456.—Diesel Four-stroke Engine.

compression stroke, air from the cylinder enters the pump, which, still further compressing the air, delivers it to the air-blast reservoir. From the reservoir the air blast is used to inject petroleum, at 49 to 55 atmospheres pressure, into the cylinder. Any surplus air or air pressure is conveyed to a starting reservoir. The makers state that the Diesel engine is always as ready and as easy to start as a steam engine ; air at about 600 lb. pressure can be admitted to the cylinder through *F* for starting purposes, and with a proper arrangement of cylinders and cranks the engine can thus be started in any position.

In Figs. 457 to 462 a series of sections and a plan of the head of the cylinder is shown with the various cams and valve mechanisms, as fitted by Messrs. Carels frères of Ghent. Fig. 457 shows the air-inlet valve, which is operated by the rocking lever shown and a cam on the cam shaft *T*. The inlet valve is not water-cooled, but the exhaust valve, which is operated by a similar mechanism, as shown in Fig. 458, is water-cooled and hollow. Water flows downward into the valve cavity through a tube fitted inside a hollow spindle, and upward through the space between the tube and hole in the spindle, to a stationary overflow at the top end. The air-starting valve *F* is shown in Fig. 459, and, its cam mechanism being fitted alongside the fuel inlet mechanism, can be thrown out of gear by a lever *P* shown in dotted lines for the starting position and in full lines in Fig. 456 for the working position. This lever rotates the shaft *J* and eccentric fulcrums, on which the fuel inlet and starting rocking levers pivot, in such a way that the starting valve is closed when the fuel valve is in action, and *vice versa*. The four rocking levers are shown in plan in Fig. 460, which, reading from the top of the illustration, occur in the following order : air inlet, starting air inlet, fuel valve with pipe connections for air and fuel respectively, and exhaust outlet with two pipe connections for inlet and outlet cooling water.

Fig. 461 shows the system of cylinder lubrication. A series of holes is drilled into the inner or working (liner) barrel of the cylinder, at the position shown in Fig. 456 and *XX* in Fig. 457 ; a groove is then cut circumferentially on the exterior of the liner, and over this groove a ring is shrunk on so as to form a channel in communication with all the holes. A connection is formed between this channel and the outside of the cylinder, and oil is forced in by a pump when the piston is on the bottom dead centre and the holes are opposite the space between the bottom piston ring and the one next above it.

Fig. 462 shows an enlarged section of the Diesel oil fuel valve and

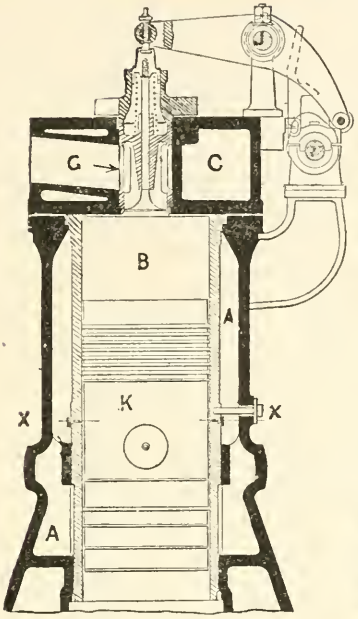


FIG 457.—Air Inlet Valve.

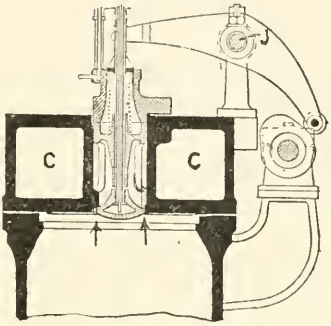


FIG. 458.—Exhaust Valve.

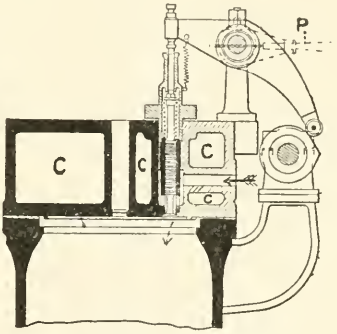


FIG. 459.—Starting Valve.

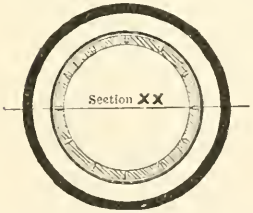


FIG. 461.
Lubrication Holes and Channel.

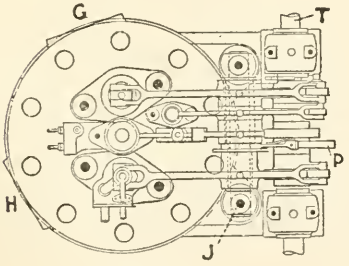


FIG. 460.
Rocking Levers and Starting Lever.

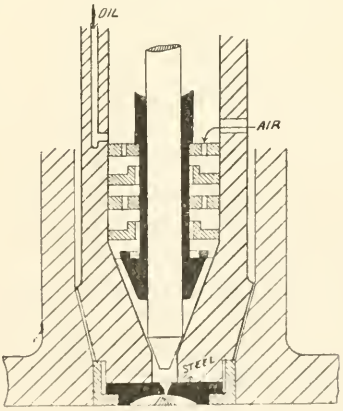


FIG. 462.
Oil Fuel Valve and Atomiser.

FIGS. 457 to 462.—Diesel Valve Mechanisms and Lubrication.

atomiser. The oil enters under pressure from the oil fuel pump through the channel shown, and is atomised by an air blast of 750 to 800 lb. pressure per square inch in passing through several (usually four) perforated plates. The holes are about eight-hundredths of an inch in diameter, and zigzagged out of line. The atomised fuel and air pass through a steel nozzle, and meeting the air in the cylinder, heated to a temperature of about 1000° F. by compression, ignition ensues, and

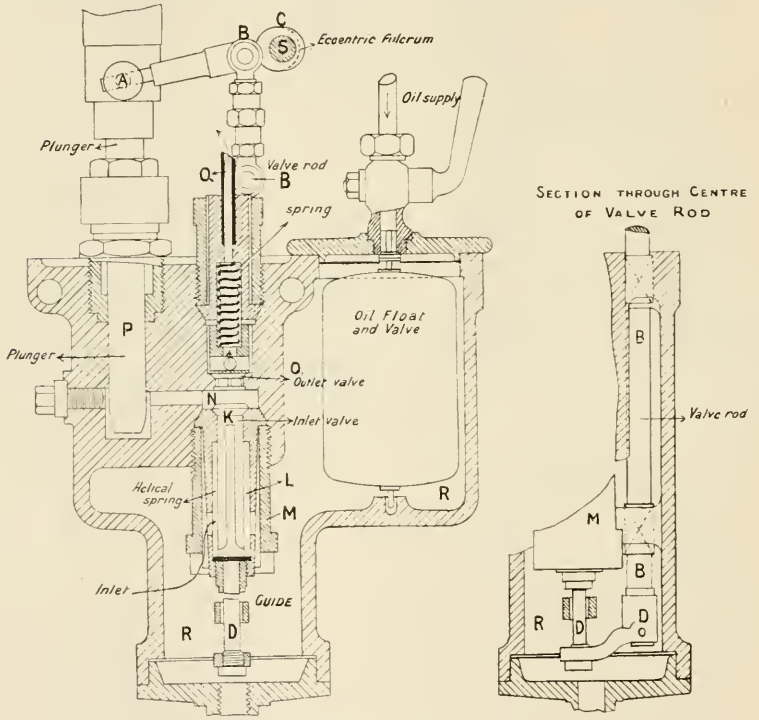


Fig. 463.—Diesel Oil Fuel Pump.

the combustion proceeds steadily and comparatively slowly for an appreciable part of the stroke, as shown by the indicator diagrams in Fig. 464. In this system it should be noticed that the oil enters the cylinder as a liquid, not as a vapour. Vaporisation takes place almost instantaneously, but the heat required to convert the liquid into vapour is supplied by the working substance and, consequently, tends to reduce the temperature of combustion. The comparatively slow combustion is principally due to this cause.

The oil fuel is injected into the cylinder by a pump, Fig. 463, the

plunger P of which has a constant stroke and is driven from the end of the valve cam shaft (T in Fig. 456). ABC is a rocking lever with a fulcrum C, and is actuated by the movement of the plunger P. The valve rod BB is worked by the rocking lever, and, by means of the arm and spindle D, opens the oil inlet valve K against the action of the spring L. The fulcrum C is fixed eccentrically on a shaft S which is caused to rotate, but not to make a complete revolution, backwards and forwards as necessary, by the action of the governor. The action of the governor alters the stroke of the rod BB, and thus controls the quantity of oil passing through the inlet valve K into the small chamber N, whence it is forced by the pump through the outlet valve O to the pipe Q and to the oil fuel valve (E in Fig. 455) in the head of the cylinder.

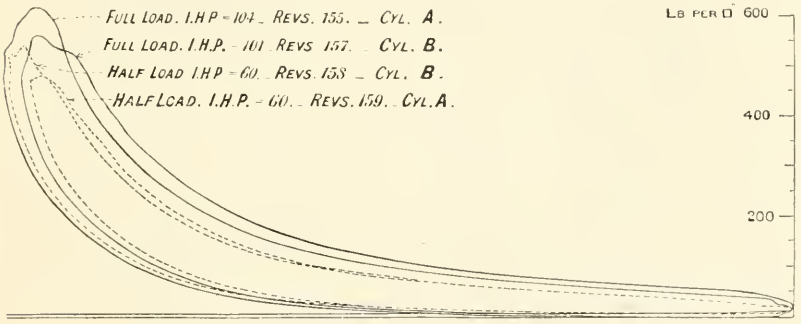

FULL LOAD. I.H.P = 104. REVS. 155. CYL. A.
FULL LOAD. I.H.P. = 101 REVS 157. CYL. B.
HALF LOAD I.H.P = 60. REVS. 158 CYL. B.
HALF LOAD. I.H.P. = 60. REVS. 159. CYL. A.
LB PER □ 600
400
200

FIG. 464.—Indicator Diagram of Diesel Engine.

The oil float automatically regulates the level of oil in the float chamber RR.

Liquid fuels differ far more in constitution than their specific gravity and flash point appear to show, and successful working can only be expected when the oil fuel atomiser, Fig. 462, is adjusted to suit the particular grade of fuel in use, and adjustment may be necessary with each change of fuel.

In Fig. 464, four indicator diagrams are shown, taken under various conditions from the two cylinders A and B of a Diesel engine of 160 Brake H.P. The principal difference between these and the diagram shown in Fig. 438 is that in the Diesel engine, although the compression is higher, there is a continued high pressure for a greater proportion of the working stroke.

Fig. 465 shows a two-stroke-cycle reversible Diesel engine as constructed by Messrs. Sulzer of Winterthur. This type is fitted with a

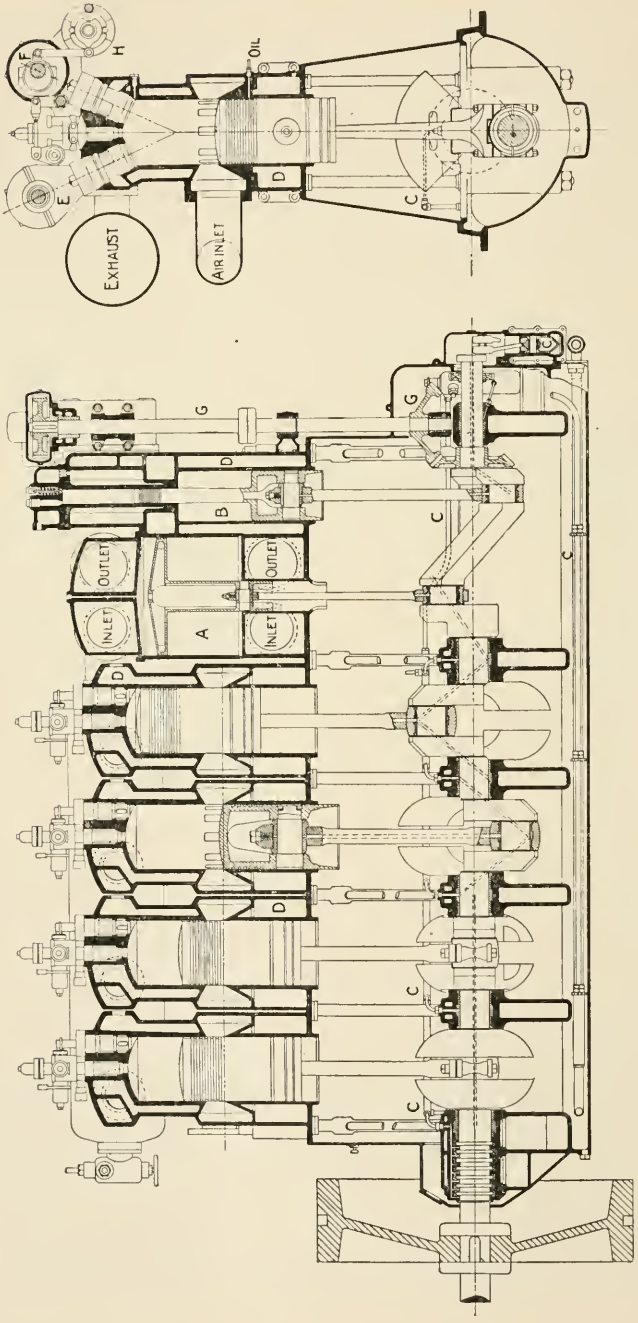


FIG. 465.—Sulzer-Diesel Single Acting Two-stroke-cycle Engine.

separate scavenging pump *A*, and a separate two-stage compression fuel injection pump *B*, which takes its supply from *A*, and delivers into an air reservoir from which the fuel injection is supplied as required. Forced lubrication *C* is fitted, and this practice is usually followed in marine types of oil engines. A pump is also fitted for circulating water through the jackets *D*. Reversing is effected through a double set of cam shafts *E* and *F* fitted above the cylinders, and driven by gearing *G* from the engine shafting ; and by means of a lever and control shaft *H*, fitted with cams, the operation of the injection valves is thrown on to either cam shaft (Ahead and Astern) as required. Starting is effected by compressed air from the reservoir as in other types of Diesel engines. Although the general arrangement is followed in more recently constructed engines, these are now almost invariably fitted with a separate guide and a connecting rod, as in reciprocating steam engines, and thus one of the worst features of unsuccessful practice is eliminated.

Diesel-Sulzer Engines of the "Monte Penedo."—The ship is propelled by twin four-cylinder, reversible, two-stroke-cycle oil engines, Figs. 466 and 467. The engines are of the crosshead type with cylinders $18\frac{1}{2}$ inches diameter and $26\frac{3}{4}$ inches stroke, and on service each cylinder develops about 210 to 250 S.H.P. at 160 r.p.m. The cylinders are carried at their upper ends, from the top of long steel columns *A* fixed in the bed-plate *B*, and are thus free to expand downwards. The covers are separate from the cylinders and are secured by the same columns *A*, and the pistons, which are hollow and water-cooled, can be withdrawn from below. The fuel and starting valves are fitted in the covers and are driven off a single cam shaft *C*. A centrifugal governor *D* controls the power of the engine by putting the fuel valves out of action one after the other in the order 1, 3, 2, 4, and they are subsequently thrown in again in succession. Compressed air is supplied for the first starting. When reversing, the engines are first stopped and then started on compressed air, and the fuel is put on successively in groups of two cylinders. The scavenging air inlet *F* is arranged on a level with the exhaust *E*, and thus back firing into the scavenging receiver is avoided. The scavenging valve is in a receiver, and therefore protected from the heat of combustion in the main cylinders.

The engine cylinders are water-cooled, the supply being drawn from and returned to the sea by pumps driven off the main engines, but when in harbour, where dirty water which also contains organic matter would be harmful, the cooling water is drawn from and sent

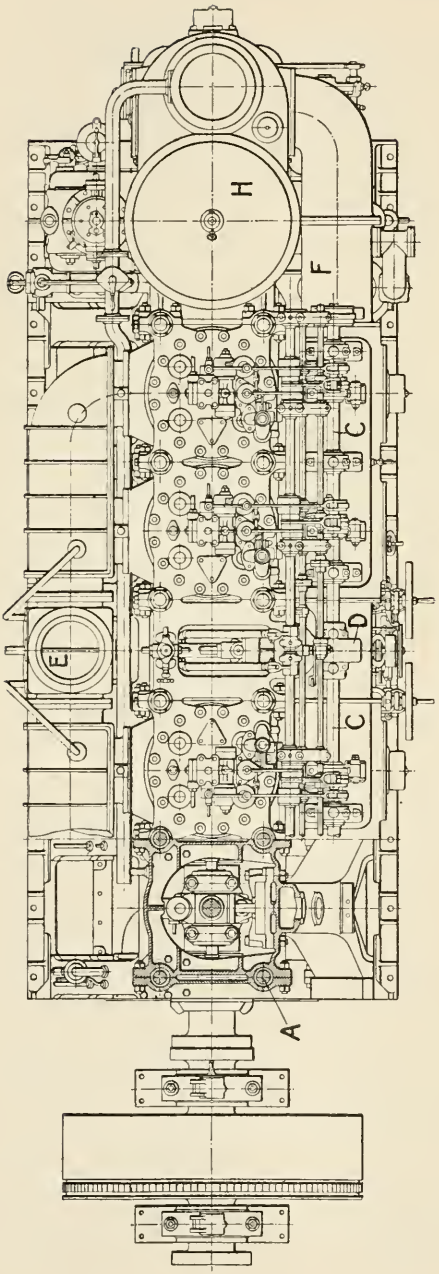


Fig. 466.—Diesel-Sulzer Engines of *Monte Penedo*.

back to a cooler which is cooled by sea water. The piston head has two pipes *G* fixed to it, moving up and down in larger pipes fixed to the frame. Another pipe fixed to the frame passes up inside one of the moving pipes conveying the water-supply; neither gland nor water packing is used.

The scavenging pump is at the forward end of each engine and forms a fifth cylinder *H*; it is double-acting and the distribution is effected by piston valves operated by a link motion *K* and eccentrics fitted on the end of the crank shaft. A three-stage compressor, driven from the same crank as the scavenging pump, supplies air at about 995 lb. pressure per square inch for injecting fuel, starting and reversing, etc. The first stage *L* is immediately below the scavenging cylinder, the second *M* and third *N* stage are driven by an arrangement of rocking levers *P*. The three-stage compressor is water-cooled in each stage.

Each engine drives direct by rocking levers, a cooling water pump, a bilge pump, a sanitary pump, a piston-head cooling pump, an oil

fuel pump, and a pump for supplying oil throughout the forced lubrication system.

A steam boiler is fitted on deck and is fired by oil on the Koerting system, and supplies steam to the steering engine (which can be alternatively driven by heated compressed air), to one 40-tons-capacity

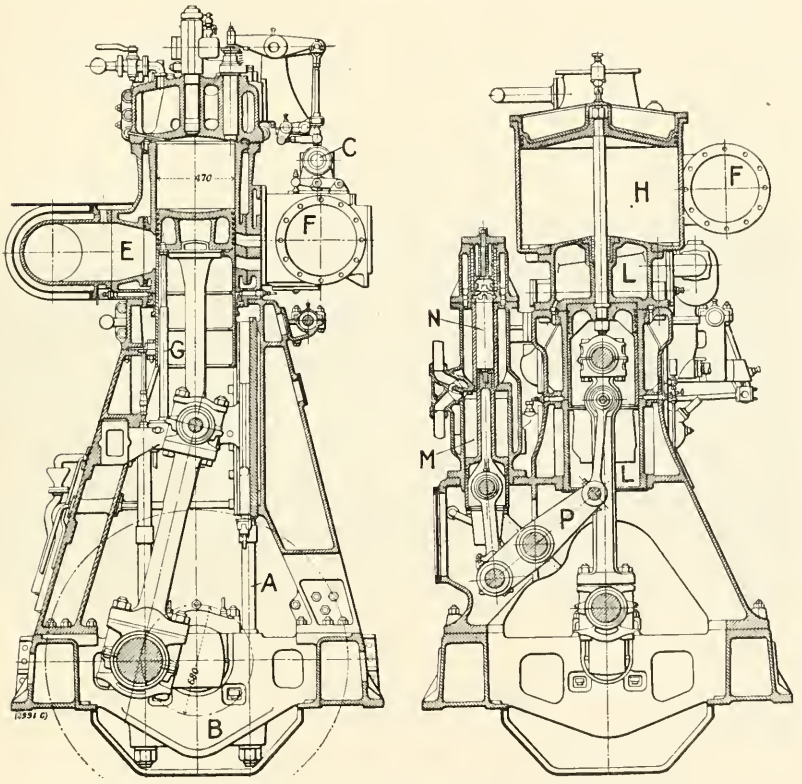


FIG. 467.—Diesel-Sulzer Engines of *Monte Penedo*.

ballast pump, and to a 10-horse-power dynamo. The other auxiliaries are a 50-horse-power Diesel-driven dynamo, a 50-horse-power Diesel-driven air compressor, and one electrically driven ballast pump of 120 tons capacity per hour.

It should be noticed that the top of the piston in the more recent types of engine is concave, and not convex as in the earlier types: also that the upper part of the cylinder is considerably thicker than the

lower part, and at the top there is a very great increase in strength and thickness in the form of a flanged ring.

Another point of importance is the necessity, with single-acting engines particularly, of a large and heavy fly-wheel. With six cylinders it might be possible to decrease the fly-wheel capacity considerably, and such an evolution is extremely desirable. With double-acting engines the fly-wheel capacity is of less importance, but at present it does not appear possible to entirely eliminate it as in marine steam practice.

Submarine Engines.—In the L class submarines, built by Messrs. Vickers, which have a submerged displacement of about 1000 tons,

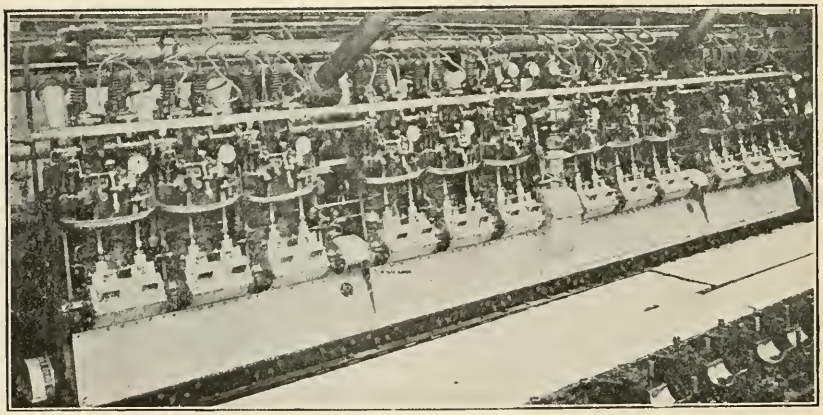


Fig. 468.—Vickers non-Reversing Submarine Engine.

the twin screws are worked by four-stroke-cycle, 12-cylinder engines developing a total B.H.P. of 2600 (Fig. 468), or rather less than 110 B.H.P. per cylinder, which gives a surface speed of about 17 knots. Electric motors for submerged running and for reversing are fitted to both shafts, and give a speed of about $10\frac{1}{2}$ knots when submerged and developing 1600 B.H.P.

About the same power is fitted in submarines by Messrs. Franco Tosi of Milan ; one set of twin screw engines is of the two-cycle type and another of the four-cycle type with eight cylinders instead of the six in the former. Nearly 220 B.H.P. is obtainable from the two-cycle type per cylinder and nearly 163 from the four-cycle type. This gives a fairly good comparison between the two types, which is given in the table following :

	Full Load.	Full Load.	¾ Load.	½ Load.	Over Load.
Type of engine . .	2-cycle	4-cycle	4-cycle	4-cycle	4-cycle
Number of cylinders .	6	8	8	8	8
Duration of trial . .	..	36 hours	96 hours	12 hours	1 hour
B.H.P.	1300	1300	950	650	1485
Revolutions per minute	300	300	260	240	..
Fuel oil consumed per B.H.P.-hour . .	0·540	0·390	0·410	0·438	..
Lubricating oil consumed per B.H.P.-hour	0·0334	0·0180	..	..	..

An illustration of the Franco-Tosi Submarine Engine of the four-cycle, 8-cylinder type is shown in Fig. 469, and some of the component parts of it in Figs. 470, 471, and 472. Each cylinder has two inlet,

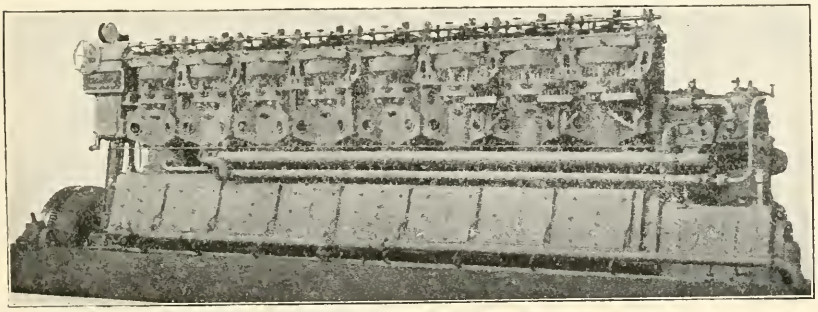


FIG. 469.—Franco-Tosi Four-stroke-cycle Submarine Engine.

two exhaust, and one fuel injection valves. The inlet valves are fitted in the cylinder head, and the exhaust valves in separate cages. The engine is not reversible, and the single cam shaft runs along the top of the cylinders, being driven from the crank shaft through a vertical shaft and skew gearing (see Fig. 465); the valves being operated by the usual roller and lever mechanism (see Figs. 457 to 462). A four-stage air compressor is driven from the forward end of the engine, which is of trunk type without piston rods.

The main trunk pistons are in two parts, the upper carrying the seven piston rings (Fig. 470), and the lower carrying the gudgeon pin and taking the side thrust of the connecting rod. The piston is oil-cooled through a spiral duct in its crown (Fig. 470).

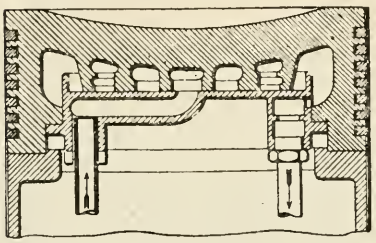
The gudgeon pin is prevented from turning by a set screw (Fig. 471), and is fitted with a sleeve keyed on to the pin and forming a bearing

surface which it thus considerably increases so as to reduce the pressure on it per square inch from 2000 to 1400 lb. White metal is run into the eye of the connecting rod and the bearing is forced lubricated through a pipe fitted inside the hollow connecting rod.

The exhaust valves (Fig. 472) are water cooled through a spiral duct and pipe attached to a special nut on the top of the exhaust valve spindle.

The main water-cooled exhaust connecting pipe is fitted internally with a light spiral baffle to prevent interference of the exhaust of one cylinder with another, since in an 8-cylinder four-cycle engine two cylinders will exhaust at the same time at certain points in the revolution.

There is no starting device in this Tosi engine, and either the main



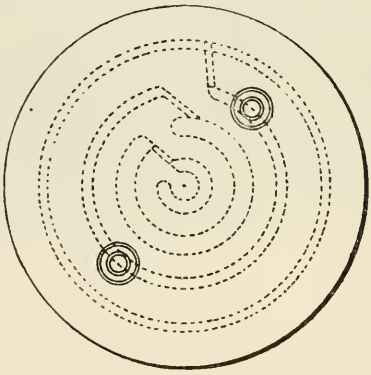


Fig. 470.—Tosi Oil-cooled Piston.

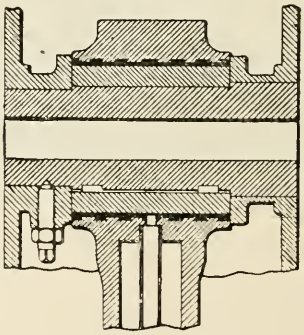


Fig. 471.—Tosi Gudgeon Pin.

electric motors or a separate starting engine can be used. Reliance on the main electric motors for starting is not always well placed because it is possible to wreck the engine by starting it full of water from submergence or other cause, and in some instances which came under the author's notice the engine was badly injured and took some weeks to repair. In one instance the wrecking was due to a faulty connection of the electric switch and tell-tale which showed ahead instead of astern; when being driven astern by an external force the water in the cylinders cannot escape through the usual channels and

thus an excessive pressure is set up in all internal parts. Divided control, as always, is a rotten system of running machinery and proved itself in this particular case.

In later types the Franco-Tosi engine is provided with a separate compressed air starting engine with three radial cylinders acting on a shaft and pinion with teeth engaging in the periphery of the fly-wheel. The teeth are slightly spiralled, or inclined, to the axis, so that when the main engine functions and becomes the driver, the thrust of the spiral teeth pushes the pinion forward and so out of engagement. To reduce the torque required when using the auxiliary starting engine, the main engine can be decompressed until the requisite speed of revolution is obtained.

For the Franco-Tosi 1300 B.H.P. engine, all the auxiliaries are driven by electric motors, including the cooling water pump and the lubricating oil pumps. The weight of the engine is stated to work out at 66 to 70 lb. per B.H.P.

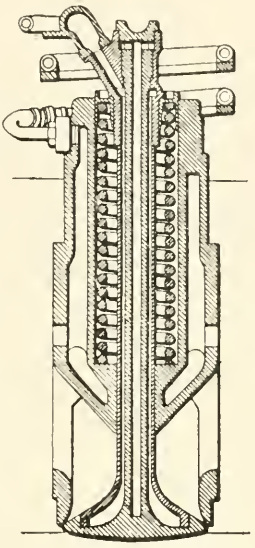


FIG. 472.—Tosi Exhaust Valve.

Motor Ship "Asia."—As a recent example, several figures are shown of the Diesel engines of this vessel which were constructed by Messrs. Burmeister & Wain, Copenhagen, for the East Asiatic Company.

(Figures 473 to 478 and extracts of description are taken from *Engineering*, January 2, 1920.)

The two sets of main engines are of the four-stroke-cycle, single-acting, directly reversible type, each having six cylinders, 630 mm. (24·8 inches) in diameter, with a stroke of 960 mm. (37·8 inches). At 125 revolutions per minute the engines are together capable of developing 3100 I.H.P., continuously, under normal conditions at sea, giving the ship a speed of $11\frac{1}{2}$ knots.

The *Bedplate* is divided into two parts bolted together near the mid-length of the engine, and the cross pieces which carry the main bearings are of box section, and the brasses are cylindrical so that the lower ones can be removed without lifting the crank shaft. An oil-trough of welded steel plates is bolted to the bottom of the bedplate, as shown in the cross section sketches, with the necessary drain pipes.

The *Standards* are in the form of A-shaped castings of box section.

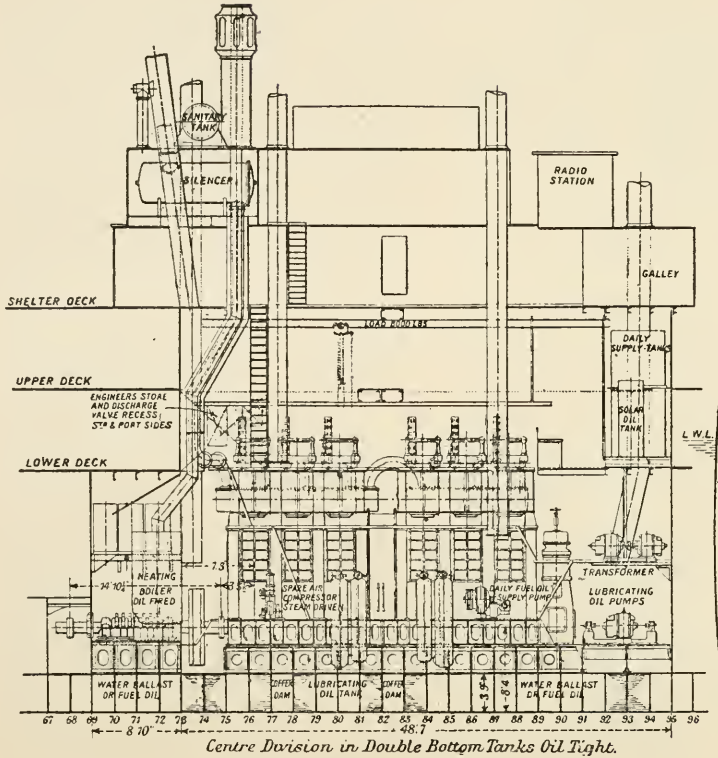

SANITARY TANK
SILENCER
RADIO STATION
GALLEY
SHELTER DECK
LOAD 8000 TONS
DAILY SUPPLY TANKS
UPPER DECK
ENGINEERS STOAL AND DISCHARGE VALVE RECESS STB & PORT SIDES
SOLAR OIL TANK
L.W.L.
LOWER DECK
HEATING BOILER OIL FIRED
SPARE AIR COMPRESSOR STEAM DRIVEN
DAILY FUEL OIL SUPPLY PUMP
TRANSFORMER
LUBRICATING OIL PUMPS
WATER BALLAST OR FUEL OIL
COFFER DAM
LUBRICATING OIL TANK
COFFER DAM
WATER BALLAST OR FUEL OIL
67 68 69 70 71 72 73 74 75 76 77 78 79 80 81 82 83 84 85 86 87 88 89 90 91 92 93 94 95 96
8 10"
48.7
Centre Division in Double Bottom Tanks Oil Tight.

Fig. 473.—Longitudinal Section.

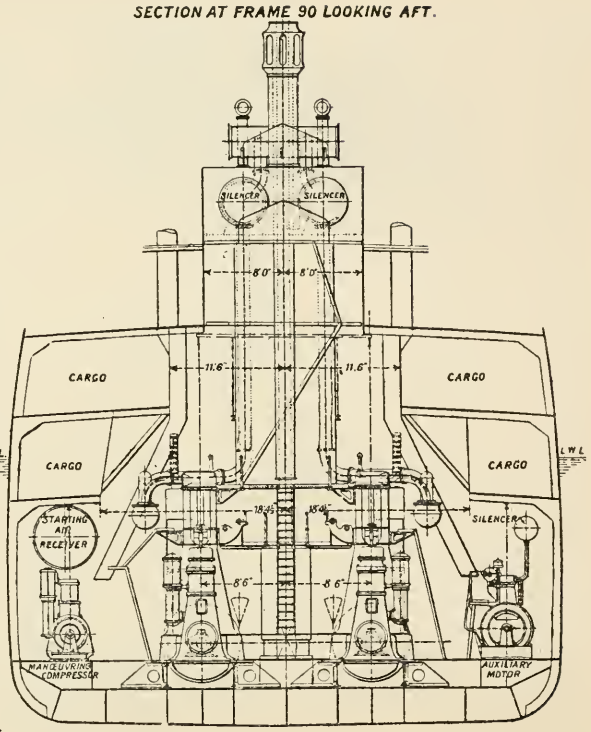


Fig. 474.—Cross Section, looking Aft.

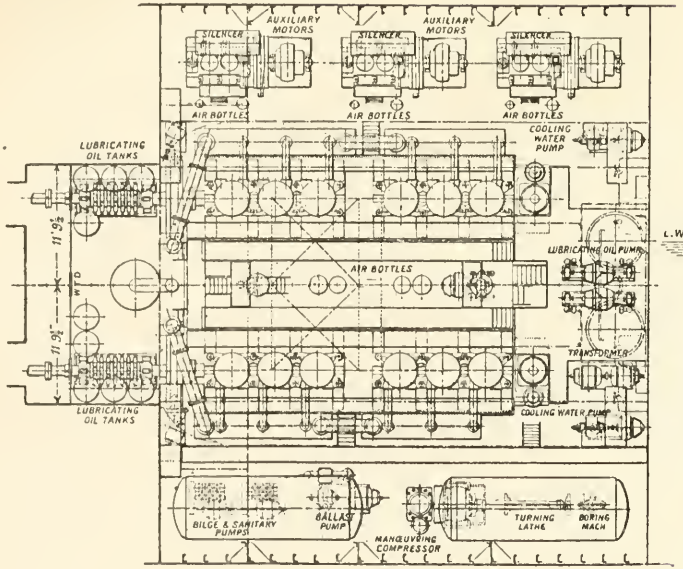


FIG. 475.—Plan of Engine Room.

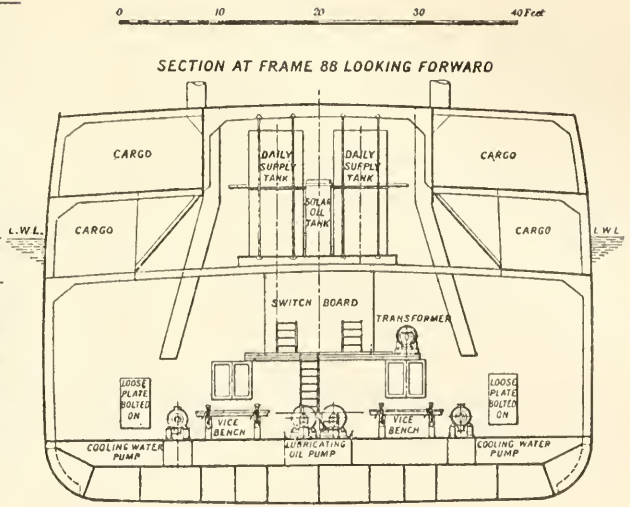


FIG. 476.—Cross Section, looking Forward.

There are eight of these standards to each engine, one being placed between each pair of cylinders and one at each end of the two groups. At the back of the engine the standards are connected by castings forming the slipper *Guides* and at the front the connection is made by hollow distance pieces (Fig. 477).

The *Guides* are provided with removable faces, and cooling water is circulated between these faces and the main casting. The spaces between the standards are closed by thin iron sheets, the edges of which are fitted with angle irons to make oil-tight joints with the standards. The tops of the standards are provided with covers, through which the piston rods pass by oil-tight bushes. The whole of the working parts are thus completely enclosed, but by removing the front doors the crossheads and crank shaft bearings can be easily inspected.

The *Crank shafts* are of the built-up type designed in accordance with the usual mercantile practice, and are made in two parts each having three cranks. The forward part has also a crank bolted on for driving the main compressor.

The *Cylinders* are arranged in groups of three, and each cylinder is fitted with a *Liner* which is kept in position by the cylinder cover and can be removed when necessary. The cylinder castings are extended to form feet, which rest on the standards, leaving a space between the covers of the standards and the end of the liner, so that the internal surface of the latter can be examined. Each group of cylinders is secured by eight long bolts, two in the space between each pair of cylinders, and two at each end. These bolts pass down through the standards, and through the crosspieces of the bedplate quite close to the main bearings (Fig. 478). They thus serve to take the whole of the upward force on the cylinder covers due to the explosion, without subjecting the standards to any tensile stress. This arrangement is similar to the Sulzer engine shown at *A* in Figs. 466 and 467.

The short *Pistons* are made with the head hollowed out under the fuel injection valve (Fig. 477). Each piston is fitted with eight packing rings and is cooled by sea-water, which enters and leaves through telescopic pipes connected with the water cooling system of the engine. The outlet is visible so that the attendant can satisfy himself as to the quantity and temperature of the water flowing at any time. Each piston is bolted directly on to a large flange formed on the upper end of the piston rod, and this method obviously diminishes the stresses on the piston due to the explosion.

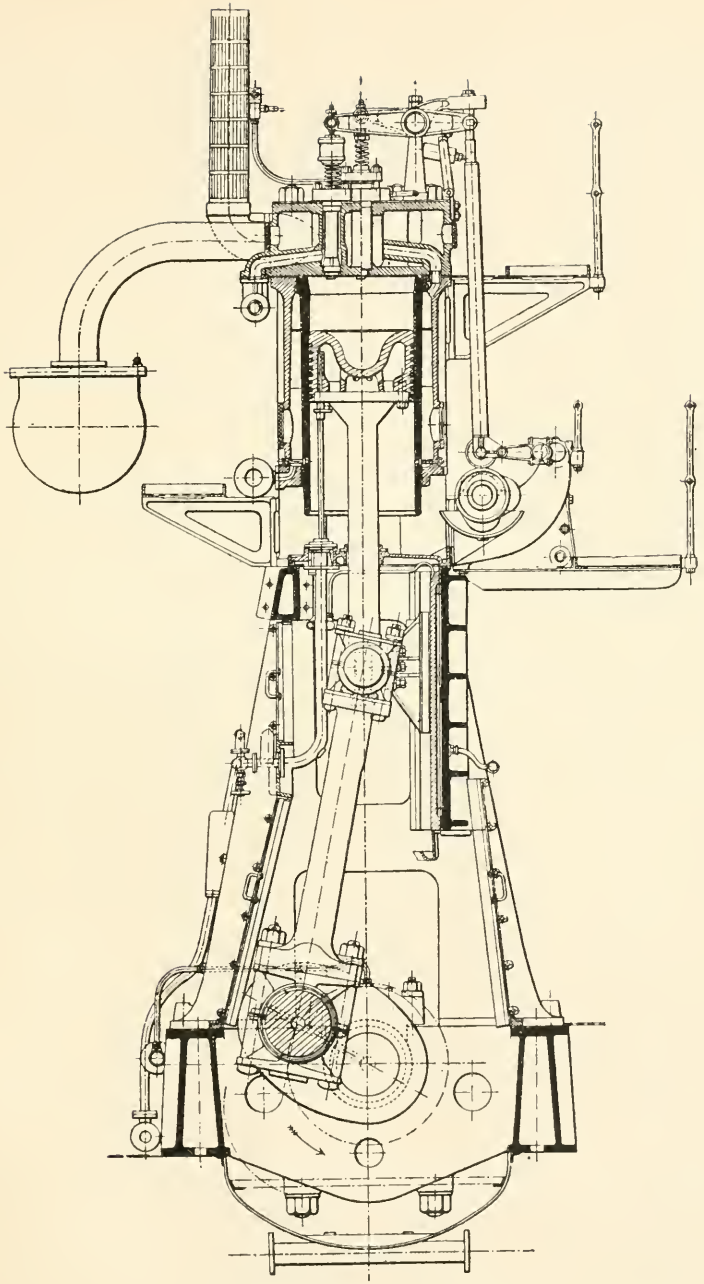


Fig. 477.—Sectional Elevation of Engine.

The *Connecting Rods* are designed in accordance with ordinary reciprocating practice. At the piston rod end, which is forked, the loose brasses are lined with white metal. The crank bearings are also lined with white metal run into steel half bushes.

In addition to the four valves shown in Fig. 460, there is a relief valve, and all five valves are fitted in the cylinder cover. The four working valves are operated by push rods and rocking levers from a cam shaft running along the back of the engine just above the tops of the standards.

The *Cam shaft* is fitted with double cams (Fig. 477), and reversal is effected by moving the shaft along horizontally, so as to bring the respective cams for the Ahead and Astern directions into contact with the rollers on the push rods.

A *Reversing shaft* runs parallel with the cam shaft, and is fitted with cranks which are connected by means of links with the push rods. When this shaft is turned, the push rods and rollers are pulled clear of the cams so that the cam shaft can be moved along. The rotation of the reversing shaft, and the movement of the cam shaft are effected simultaneously by the movement of a sliding block in a nearly vertical direction. The block carries a cam, which engages with the cam shaft, and is also fitted with a rack which gears with a toothed wheel on the reversing shaft. This cam is straight at both ends and curved in the centre, so that the first part of the motion of the block only rotates the reversing shaft and thus pulls the rollers clear of the cam shaft. Afterwards the curved part of the cam on the block engaging with the cam shaft, moves the latter along, and after this has come to rest in its new position, the rollers are replaced on the cams by the continued rotation of the reversing shaft. The sliding block is moved by means of a direct acting cylinder and piston operated by compressed air, supplied from the starting air vessels of the engine. This arrangement is similar in principle to Brown's reversing gear (Fig. 209).

The cam shaft is driven by a train of spur wheels (Fig. 478) which are completely enclosed in the crank casing, and the bearings are supplied with forced lubrication, as are also all other bearings similarly enclosed.

The *Fuel oil pumps*, one of which is provided for each cylinder, are driven by eccentrics mounted on a shaft on which one of the spur wheels of the cam shaft driving train is keyed.

Governors are also fitted, and these are driven from the same shaft as the fuel pumps.

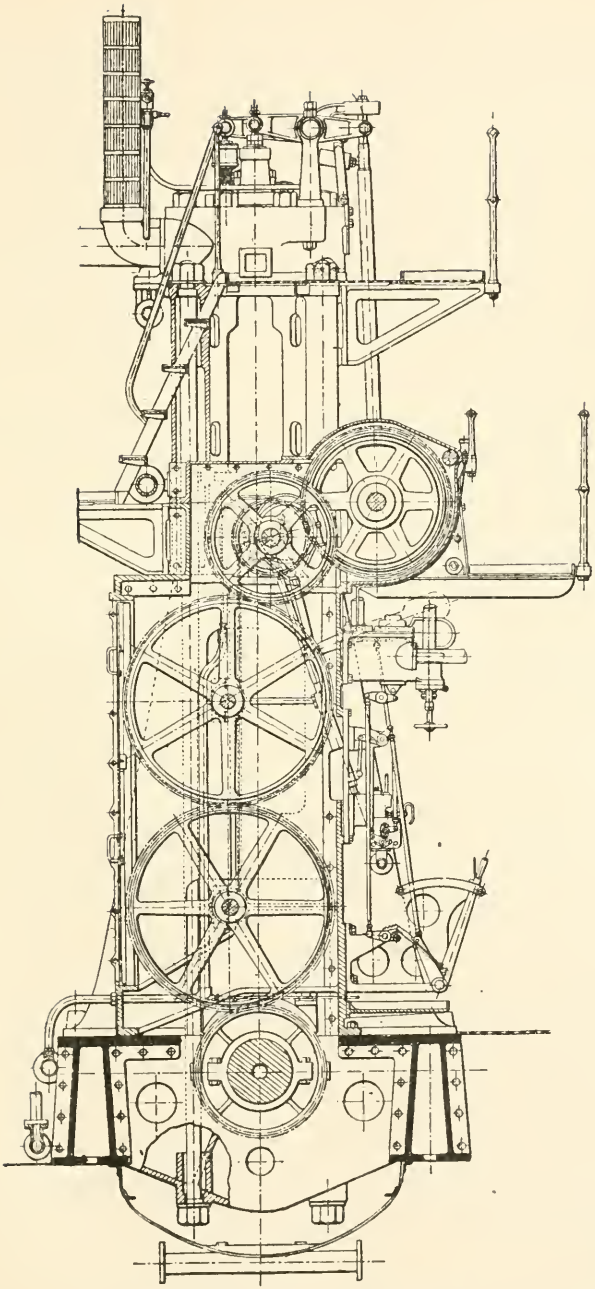


Fig. 478.—End View showing Reversing Gear.

Compressed air is supplied for *Fuel injection* by a three-stage compressor mounted on the forward end of the engine, and driven by a double crank shaft with connecting rod and crosshead, and the cylinders have been completely separated from the crank casing of the engine to prevent the possibility of oil being drawn in with the air. The low pressure, and intermediate pressure, coolers consist of cast iron casings with tube plates and straight brass tubes arranged similarly to a marine condenser, except that, owing to the high pressure used, the tubes are expanded into the tube plates. The high pressure cooler is a copper coil enclosed in the cooling jacket surrounding the high pressure cylinder. Automatic valves are employed throughout.

The *Thrust block* is of the ordinary horse-shoe type, and is mounted separately from the engine. A *Flywheel* is fitted on the thrust shaft between the block and the engine, and a worm-barring gear, operated by an electric motor, is also provided (Figs. 473, 475).

The engines are started by air at a pressure of 25 atmospheres on the builders' patented system, in which the starting valves are put into or out of operation by means of small air cylinders. The same handle is used for starting as for control of the fuel pumps and for regulating the speed of the engine. The system thus approximates to that used for an ordinary reciprocating steam engine, when starting and reversing.

All the auxiliaries throughout the vessel are operated electrically, except the main engine fuel oil pumps and oil fuel injection pumps already mentioned above. Current for working the auxiliaries, and also for lighting the ship, is produced by three 220-volt continuous current generators directly coupled to Diesel engines, each of which has two cylinders and is capable of developing 90 B.H.P.

A two-stage compressor is used to supply air for manœuvring the main engines ; this auxiliary serves as a spare for one of the main compressors, in accordance with Lloyd's requirements. There is also a steam-driven air compressor for use after the ship has been laid up or for emergency use in case no air is available from the air bottles. In this case the bottles can be charged for starting the auxiliary Diesel engines, which can then be used for starting the main engines. Steam for the small compressor is supplied by a donkey boiler (Fig. 473), mounted in the space between the thrust blocks, and fired by oil fuel atomised either alternatively, by steam, or by compressed air.

Two rotary pumps of the gear wheel type are used for supplying

oil for *forced lubrication*, which is fitted throughout the main engines and small Diesel engines.

There are two settling tanks, each sufficient to hold enough fuel for 12 hours' working ; these tanks are filled alternately and allowed to settle for as much time as possible to clear dirt and water.

On a stated consumption of 10·4 tons per 24 hours at sea for all purposes, the capacity of the fuel tanks of 1232 tons gives a range of 33,000 sea miles without refuelling. The fuel consumption works out at 0·306 per I.H.P. or 0·397 per S.H.P. hour.

Electric-driven hydraulic variable speed gear is fitted for steering the ship.

Opposed Piston Engines.—Internal combustion engines of this type are under construction by Messrs. Cammell Laird and other firms.

There are several ways of arranging the mechanism for marine propulsion, and one is illustrated in Fig. 149c for an ordinary steam engine which could also be adapted to the internal combustion engine and has been used in this way.

In any method adopted for internal combustion, it is in every case necessary to fit some definite arrangement of rods or levers to connect the cranks so that the inward end of the working strokes of the opposed pistons shall exactly synchronise to effect sufficient compression to produce the pressure necessary for ignition ; and therefore the motive power without such synchronism will not be continuous or operative unless the necessary connection is made between the cranks or rods.

The great advantage of the opposed piston type is the avoidance of extensive staying of the cylinder covers or heads, as shown in Figs. 466, 467, and 477. The pressure is conveyed directly in the opposed piston type to the working parts and not through the framing. To obtain the necessary small space between the two opposed pistons for the required high compression, the ends of the piston are usually made flat or convex instead of concave, as in the usual type shown in figures above mentioned. Between the two pistons, space must be allowed for fitting the inlet and starting holes in connection with the oil fuel and air reservoirs for a two-stroke-cycle engine ; and in a four-stroke-cycle engine space must also be allowed for the exhaust ports.

The necessary levers or cams for operating the various valves can be fitted to work from the cam shaft in connection with the crank shaft as in other oil engines.

With a well-designed engine the opposed piston type gives every

chance of producing no sensible vibration from reciprocation of the pistons and other working parts of the engine, and a fairly equable turning moment can be obtained for each unit of two cylinders. By increasing the number of units the turning moment is almost flat.

One form of the *Cammellaird-Fullager* opposed type of engine is illustrated in Fig. 479. (Taken from *Engineering*, January 30, 1920.)

Each unit consists of four pistons, of which *A* and *B* work in one cylinder and *C* and *D* in another close alongside the first. By means of crosstie rods the piston *A* is connected with the crosshead of *D* and the crank *F*; and the piston *C* is similarly connected with the crank *E*. The cranks *E* and *F* are opposite each other at an angle of 180 degrees.

The engine is arranged to work on the two-stroke cycle and the pistons *A* and *B* are shown at the beginning of the working stroke, during which the crank *E* will be pushed downwards by the piston *B*, and the crank *F* will be pulled upwards by the piston *A*. At the same time the piston *D* will be pushed upwards and the piston *C* downwards, and the mutual action of *C* and *D* will compress the air left in the cylinder, after scavenging is completed and the ports closed, during the inward stroke until at the predetermined point oil fuel is admitted and ignition takes place in readiness for the outward and working stroke of *C* and *D*.

The piston rods are each provided with a crosshead and guide for receiving the thrust caused by the oblique action of the crosstie rods between *A* and *D*, and *B* and *C* respectively, as shown in the illustrations. It should be noticed that these crosstie rods are in tension during the working stroke and in compression during the compression, or return, stroke.

The upper crossheads are closed in and used to compress the air to about 3 lb. per square inch pressure for scavenging, and subsequently for filling the cylinders preparatory to compression. In the position shown in the left hand figure for pistons *C* and *D*, the scavenge air, passing from the scavenge pump in the direction shown by the arrows, clears the working cylinder from the lower to the upper part of the products of combustion and leaves it full of practically clean air. For a two-stroke-cycle engine this system with opposed pistons is obviously more efficient than in the single piston type, shown in Fig. 440 or engine of similar construction, in which the scavenge inlet and outlet are on opposite sides of the cylinder and where the upper part of the cylinder forms a possible and probable pocket for part of the products

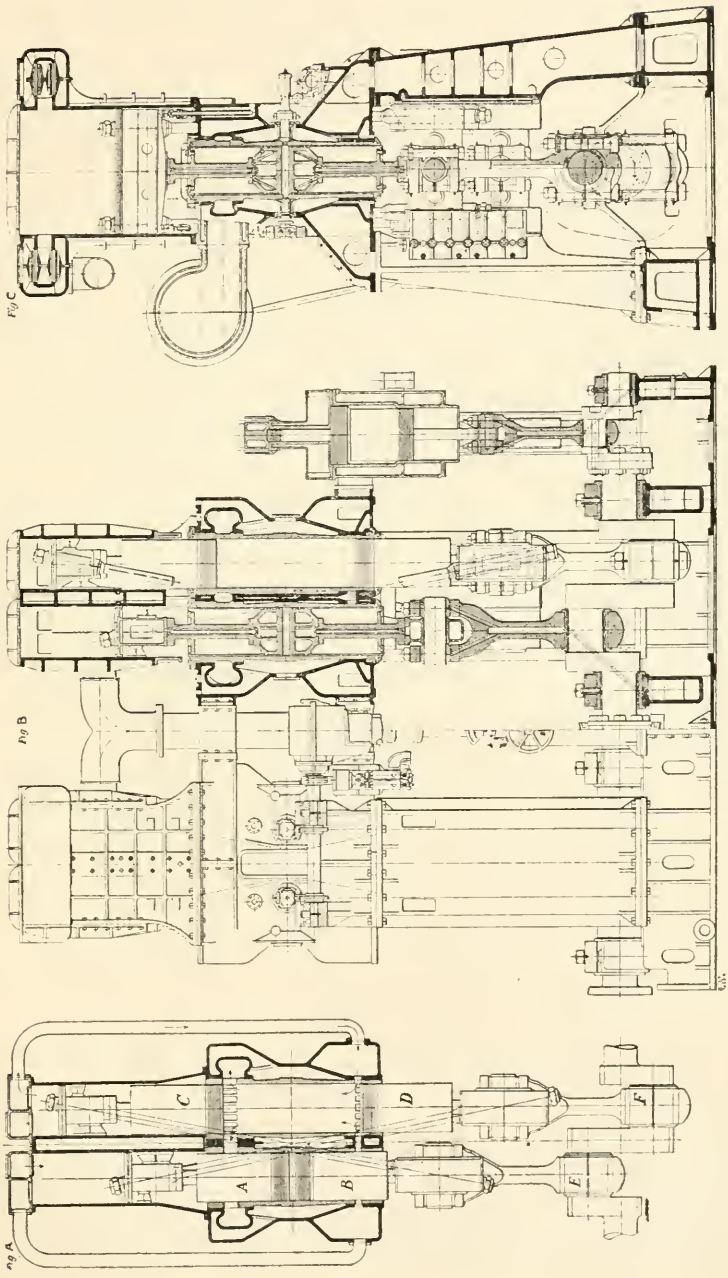


FIG. 479.—Cammellaird-Fullager Opposed Piston Engine.

of combustion and therefore reduces the effective action of the working stroke.

In the right hand figure, the inlet and outlet valves for the scavenge pump are clearly shown, and the corresponding ports are shown in sectional portion of the central figure. This figure also shows the three-stage air compressor driven by a separate crank on the main shaft for supplying air for fuel injection. The cam shaft is driven by spiral gearing from the crank shaft at the mid-length of the engine, and each cylinder is supplied with oil fuel from a separate pump driven by an eccentric on the crank shaft.

Starting is effected by compressed air supplied from a reservoir in the usual manner, and the starting valves are operated by pilot valves from the starting gear and reversing is simply effected. The starting air and oil fuel valves are provided, as shown in the right-hand figure, at the mid-length of each working cylinder, and a relief or safety valve is incorporated with the air starting valve.

Forced lubrication is fitted for the bearings, and the pistons are fitted with the usual cooling arrangements. The working cylinders are water cooled through the spaces shown.

The engine is designed to develop 1000 B.H.P. at 110 revolutions per minute, with cylinders $18\frac{1}{2}$ inches diameter and a stroke of 25 inches.

The Still Engine.—The fuel consumption of an oil-fired boiler and geared turbine installation for marine propulsion is about 0·7 lb. per S.H.P.-hour, and of a heavy oil engine working on the four-stroke cycle is about 0·4 lb. per S.H.P.-hour, in round figures. Each S.H.P.-hour represents an effective return of 2538 b.t.u. of the heat value of the fuel, whose lower calorific value may be assumed as about 19,035 b.t.u. per 1 lb.

Based on the S.H.P., the thermal efficiency of the

$$\text{Geared turbine installation} = \frac{2538}{19035 \times 0{\cdot}7} = 19 \text{ per cent about}$$

$$\text{Heavy oil engine} = \frac{2538}{19035 \times 0{\cdot}4} = 33{\cdot}33 \text{ per cent.}$$

The object of the Still engine is to utilise the heat otherwise wasted in the discharge to the atmosphere of the products of combustion, and in the jacket water discharge which together account for the loss of about 60 per cent of the heat supplied by the fuel. The Thermal Cycle

of operations of one type of Still engine is shown in Fig. 480, and it can be applied to any type of internal combustion engine working on any stroke cycle.

Referring to the figure, the upper part of the working cylinder which is the internal combustion part, is surrounded by a steam and water jacket; the lower part, which is the steam part, is surrounded by a steam jacket in which there is fitted a slide valve for admitting

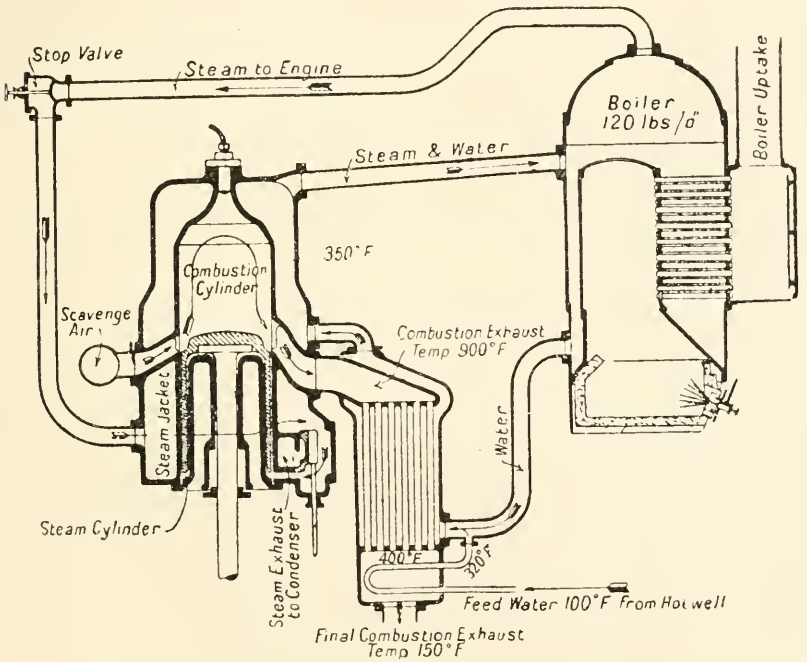


FIG. 480.—The Still Thermal Cycle of Operations.

steam to the cylinder, and exhausting the steam to a condenser or the atmosphere.

When *starting*, steam can be supplied from an oil-fired boiler, but when the engine is in normal work the oil fire can be put out, and the boiler then acts as a thermal tank to which heat is supplied by the working of the internal combustion engine. As the steam engine exhausts to the condenser or elsewhere the loss of water must be made up by a supply from a hot-well or elsewhere by a feed pump; apart from radiation, etc., the heat loss is confined to the exhaust steam waste, and the heat required for working the feed pump, circulating

pump, and, if fitted, the air pump. The sum of the work done and heat required for working these pumps may amount to about 7 per cent of the heat supplied as steam to the Still engine.

The products of internal combustion or exhaust gases at a temperature of about 900° F., pass through a number of tubes which communicate their acquired heat to water circulating from the bottom of the boiler or tank into the lower part of the combustion cylinder jacket, which is thus maintained at nearly a constant temperature of the order of 350° F., corresponding to a steam pressure of 120 lb. per square inch, and this water absorbing heat from the combustion cylinder returns to the tank as steam and water.

After passing through the tubes the exhaust gases are used to heat the incoming cold feed water, and finally escape at a temperature of about 150° F. In a four-stroke-cycle engine there is one working combustion engine stroke and two working steam strokes in every two revolutions. In a two-stroke-cycle engine each stroke is a working stroke, and the engine is virtually double acting.

The combustion cylinder being surrounded by a nearly constant steam and water pressure of say 120 lb. per square inch, and being thus maintained at a nearly constant temperature of 350° F., the fresh air entering the cylinder is warmed, and instead of the high compression necessary for ignition of the fuel a much lower pressure is required to obtain the ignition temperature ; this pressure is about 300 lb. per square inch under normal full power working. Also during the working stroke, heat is extracted from the jacket which tends to maintain a higher working pressure throughout the stroke, and consequently a higher mean pressure is obtainable in the combustion cylinder. As the maximum working pressure is less and the external pressure, exerted by the jacket, is appreciable, the thickness of cylinder or liner may be considerably less for the diameter compared with the heavy oil engine, and although the thickness is only one-third to one-fourth that in the ordinary heavy oil engine the Still engine is stated to have had no cylinder failures. There would thus appear to be no reasonable limit to the size of working cylinder which may be adopted with safety for the Still engine, and the possibilities of great power per cylinder are obvious, with all the great advantages incidental to large cylinders of great power.

In the steam cylinder heat is supplied by the steam jacket, and consequently the mean steam pressure is raised, compared with an unjacketed cylinder and ordinary steam practice, and the exhaust

steam leaves the cylinder in either a dry saturated or slightly superheated condition. The steam supply both to the cylinder and the steam jacket is derived from heat, otherwise wasted, obtained by internal combustion.

Thus, during the working strokes of both combustion and steam cylinders the thermal conditions are highly efficient.

In the Still engine, the limits of temperature in the combustion cylinder, and the final exhaust after heating the feed water may be from 2000° down to 150° F., which would give, based on the S.H.P., a thermal efficiency of about 75 per cent. In the steam cylinder the limits might be 350° and 120° for a condensing engine, and the thermal efficiency would then be about 28 per cent, but the contingent auxiliaries must be worked to obtain this result, and a deduction should be made in any estimate of this kind for these particular purposes. If the engine is non-condensing and exhausts to atmosphere or at atmospheric pressure, the efficiency would be about 17 per cent ; there are various methods which suggest themselves for improving this efficiency by fitting some automatic system of feed-water heating.

The quantity of steam made available from otherwise " waste heat " by using the Still cycle of operations is stated to be about 7 lb. per S.H.P.-hour in a four-stroke engine at full power, and ranging down to very little at light loads when the balance only just covers radiation, etc., losses. The measure of useful efficiency is thus dominated by the useful work done by the steam, which is unlikely to exceed one-third of a S.H.P. on the 7 lb. basis above stated ; this points to a possible increase in the total thermal efficiency from 33·33 per cent to about 44 per cent, or equal to about two and a quarter times the efficiency of a geared turbine installation supplied with steam from boilers fired by oil fuel of the same calorific value.

Some actual tests of a 5-inch-by-5-inch single cylinder four-stroke petrol engine on the Still cycle are given by Mr. F. Leigh Martineau in *Engineering*, June 13, 1919, which distinctly bear on this point of efficiency.

	Per cent.
Heat appearing as combustion I.H.P.	32·0
Heat appearing as combustion B.H.P.	25·0
Heat appearing as friction	7·0
Heat passed to jackets	28·0
Heat passed to exhaust	40·0
Total	100·0

	Per cent.
Mechanical efficiency	78·0
Heat finally lost to chimney	6·0
Heat finally lost as radiation	4·0
Total heat passed to boiler	68·0
Boiler efficiency	85·0
Heat to jacket and exhaust steam	58·0
Heat appearing as steam I.H.P.	8·0
Efficiency of steam engine alone	13·8
Efficiency of steam engine and boiler combined	11·8
Total brake efficiency of the engine	33·0
Percentage of gain due to steam	31·0

A fuel consumption of 0·32 lb. oil per B.H.P.-hour has been obtained experimentally, and a single cylinder engine developing 400 B.H.P. at 120 revolutions per minute, diameter 22 inches, and stroke 36 inches, is under construction by Scott's Shipbuilding and Engineering Co. with a view to building a six-cylinder engine of 2400 B.H.P.

Some comparative tests of small engines of the Sulzer and Still types have also been made by Messrs. Cammell Laird.

QUESTIONS FROM EXAMINATION PAPERS

1. What is the generally accepted explanation of the nature of heat and temperature ?

Define the "Specific Heat" of a substance. If the specific heat of air be ·2375, how many cubic feet of air can be raised one degree in temperature by one British thermal unit, the specific volume of air being 12·38 cubic feet at 32° F. ? *Ans.* 52·13 cubic ft.

2. Mention the ways in which heat is transferred from one point to another in a body, and illustrate your answer by reference to the transference of heat from the furnace of a boiler to the steam formed in it.

3. Define the term "Total Heat of Formation" of steam at one temperature from water at another temperature.

Obtain numerical results for the total heat of formation of steam from water at 100° F., at

(1) 380° F. (1) *Ans.* 1128 b.t.u.

(2) 160° F. (2) *Ans.* 1062 b.t.u.

What is meant by the dryness fraction of steam, and how would a dryness fraction of 0·9 affect your answer to (1) above ? (3) *Ans.* 1043 b.t.u.

4. For any type of Vertical Inverted Engine with which you are acquainted, sketch and describe two of the following detail parts :—

(*a*) A large piston rod, stuffing-box with metallic packing ;

(*b*) The piston-rod slipper and guide ;

(*c*) A piston slide-valve ;

making special reference to the provision made to allow for the wear of parts.

5. Define the term "the Latent Heat of Steam."

20,000 lb. of steam per hour are discharged from an engine to a surface condenser and there condensed to water at a temperature of 100° F. Find the number of lb. of circulating water initially at 60° F. that must pass through the condenser per hour in order that the temperature of the condensing water on leaving the condenser may be 85° F., the latent heat of steam at 100° F. being 1045 b.t. units.

6. State the three ways by which heat may be transmitted from one body to another, explaining clearly how the methods differ.

In a boiler, the heat from burning coal eventually appears as heat in the steam leaving the boiler. By which of the above methods has the heat been transmitted from the coal to the steam ?

7. By means of line sketches indicate how a large ship is steered by working a steering wheel on the upper deck, and sketch and describe in detail the

"differential gear" which causes the motions of the steering engine to follow those of the steering wheel.

8. Sketch and describe any type of water-tube boiler suitable for a large ship, naming the type you choose.

9. For what reason is a vacuum maintained in the condenser of a steam engine? Sketch and describe the pump by which this vacuum is produced, stating the various methods by which these pumps are driven.

10. A vessel has a bunker capacity of 1080 tons of coal. When steaming at 15 knots she burns 170 tons per day. Find approximately the quantity of coal burnt per day at 10 knots; and if the full speed of the vessel be 19 knots, find the number of nautical miles she can steam at full speed, starting with bunkers full.

11. By means of sketches show the positions of the common D slide valve relatively to the cylinder ports, and the position of the piston in the cylinder (a) when admitting steam, (b) when cutting off steam, (c) when the piston is at one end of its stroke. Mark the directions of motion of both the valve and piston on each sketch.

12. Define the terms "pitch ratio," "pitch angle," "projected area," and "slip," as applied to screw propellers.

Obtain a formula expressing the percentage of slip in terms of the pitch, revolutions of the propeller, and actual speed of ship.

13. Sketch in plan the engine room of a large ship, showing in outline the main and all the principal auxiliary engines that would be situated in it. Name each engine and state briefly its use.

14. Give an account of the addition of heat to water, and formation of steam under constant pressure. Define the terms—

Volume per lb. of steam;

Relative volume of steam;

and quote the results for atmospheric pressure.

15. Describe the construction of an ordinary surface condenser, giving details of the jointing of the tubes in the tube plates, and the tube plates to the shell.

16. Sketch and describe a single-acting air pump as usually fitted in connection with the main engines of a ship, showing clearly the various sets of valves and the means of working the pump from the main engines. What advantages has a separately driven pump for this purpose?

17. Define a "British thermal unit," and state what is meant by its "mechanical equivalent."

An engine uses 20 lb. of water per I.H.P. per hour, the feed temperature being maintained at 100° F., and the boiler temperature at 360° F. What percentage of the heat taken up in the boiler is converted into work, the total heat of steam at 360° F. being 1190?

18. Solid bodies expand generally when their temperature is raised; describe an experiment to illustrate this phenomenon, and mention two instances in which provision is made to allow for this in the propelling machinery installation of a ship, giving details.

19. Define work, and power, and investigate the formula for the indicated horse-power of an engine—with especial reference to double-acting and single-acting engines—

$$\text{I.H.P.} = 2\frac{\text{P.L.A.N}}{33{,}000}.$$

A steam cutter has a two-cylinder simple engine. Diameter of cylinder, 6 inches; stroke, 6 inches. If the mean effective pressure be 50 lb. per square inch, and the revolutions be 400 per minute, estimate the I.H.P. developed.

20. Sketch two of the following details:—

(*a*) A section through an air pump;

(*b*) A section through a boiler stop valve;

(*c*) A crosshead bearing;

giving a brief description of the parts as illustrated by your sketches.

21. Describe, with sketches, the detail of fitting of the screw propeller to the stern shaft, showing in *outside* view only the relative position of the stern bracket bearing.

22. It is generally assumed that the coal consumption for a ship is proportional to the I.H.P. developed by the engines; show that then the coal consumption for a certain voyage will vary as the square of the speed at which it is made.

23. State how you would take an indicator diagram and work out the mean pressure from it.

What are the cylinder constants?

24. Describe in general terms the feed-water arrangements for the boilers of any ship you know, stating how losses are made good and storage provided for, and detailing the pumps, suctions, and deliveries in connection with the system.

25. Describe, with sketches, the furnace fittings of any boiler with which you are acquainted; and explain how the rate of combustion of fuel on the fire-grate can be varied.

26. By means of an outline sketch show the arrangement of parts of a return tube cylindrical boiler. Letter the various parts, and give the corresponding names.

27. Describe, with detail sketches, any two of the following:—

(*a*) Method of fitting plain and stay tubes in a cylindrical boiler;

(*b*) Method of staying the front to back plate in the steam space and in the space between the tubes and furnace;

(*c*) Method of staying the top plate of a combustion chamber;

(*d*) Method of attaching the furnace to the combustion chamber and boiler front plate.

28. Why is it necessary to have a guide for a piston-rod crosshead? Show how a separate guide is required for ahead and astern motion.

29. Describe and sketch one of the following:—

(*a*) The method employed to enable a piston to work steam-tight in a cylinder;

(*b*) The stuffing-box and packing for a large piston rod.

30. If the conditions of working a turbine were initially saturated steam, temperature 380° F., and finally steam of temperature 120° F. and dryness 0·75, how much heat is absorbed by the turbine per lb. of steam?

31. Sketch three of the following details of boilers:—

(1) The connections of the elements of the Yarrow boiler with the superheater and steam drum;

(2) The supporting frames and casings of the Babcock and Wilcox boiler;

(3) Furnace of a Thornycroft boiler arranged to burn oil fuel;

(4) The tube end connections to drums of the Thornycroft boiler.

32. Describe Howden's system of forced draught, and in *outline* only sketch the arrangement with any boiler with which you are acquainted.

33. Show, by sketches, how you would arrange the furnaces and combustion chambers of a double-ended boiler, and explain how the different parts of the combustion chambers are stayed.

34. With the aid of a sketch, show what is the usual position of the working water-level in a cylindrical boiler, and explain the probable effect if the water-level is (*a*) much too high, (*b*) much too low.

35. What is meant by the statement that "the vacuum is 25 inches, while the barometer shows 30 inches"?

Assuming the pressure of the atmosphere to be 15 lb. per square inch, what is the pressure corresponding to 25 inches of vacuum? *Ans.* $2\frac{3}{4}$ lb. abs.

36. How are the tubes in a cylindrical boiler fitted? What means are taken to protect the combustion chamber ends?

37. What is meant by the *heating surface* of a boiler? Name the parts of a cylindrical boiler which form the heating surface.

38. Describe, with the aid of a sketch, the construction of a Babcock boiler, indicating the course which the feed water has to take.

39. Describe the path of the water from the feed pump of a Babcock boiler (economiser type) to its exit as steam into the steam pipe; also the uses of the various parts through which the water passes *during* its conversion into steam.

40. What are the distinguishing features of water-tube boilers as a class? What advantages are claimed for any one of them.

41. Name the various kinds of water-tube boiler of which you have any knowledge, dividing them into *curved*-tube and *straight*-tube boilers, and describe any example of the former class.

42. Explain how a boiler is liable to suffer from undue haste in raising steam, and describe the precautions that are necessary when steam is being raised. Mention the various mountings on a boiler, and their uses.

43. Describe the construction and use of the hydrometer, and state how the service hydrometer is graduated.

If two pints of sea-water are boiled until one pint remains, what is the density of the remaining water on the service hydrometer? *Ans.* 20°.

44. Describe the construction and action of the ordinary steam-pressure gauge.

45. Describe the closed stokehold system of forced draught.

How is the air pressure measured?

46. An engine overcomes a resistance of 44,000 lb. through a distance of 1500 feet in $1\frac{1}{2}$ minutes: what horse-power is represented by this performance of work? *Ans.* $1333\frac{1}{3}$ I.H.P.

47. What is the object in cutting off the supply of steam to a cylinder before the end of the stroke? Calculate the mean forward effective pressure in a cylinder using steam of 200 lb. pressure per square inch by gauge, if the cut-off is at one-third stroke and the back pressure by gauge is 90 lb. per square inch.

Ans. 50 lb.

48. Compare the advantages of a triple expansion engine with a three-cylinder engine of the same power, each of whose cylinders takes steam directly from the boilers.

49. Steam of 160 lb. absolute pressure is admitted to a cylinder and cut off at quarter-stroke: find the average effective pressure for the stroke, the back pressure being 10 lb. absolute. *Ans.* 88 lb. per square inch.

By reference to the above example, show the advantage gained by using steam expansively.

50. Steam entering a cylinder is cut off at three-quarter stroke, the final pressure is 150 lb. per square inch absolute: what is the initial pressure (*a*) by gauge, (*b*) absolute? *Ans.* 185 lb., and 200 lb.

51. What are the two important properties of steam which are made use of in connection with a set of marine engines? Explain how they add to the efficiency of the engines.

52. Describe and give an outline sketch of the "all round" variety of link motion.

53. What are the advantages of corrugated furnaces? Describe how you would fit this form of furnace to the combustion chamber.

54. Describe how an asbestos-packed cock is repacked. Sketch one so fitted.

55. Sketch a safety valve for a high-pressure boiler, showing in detail the lifting gear. What amount of lift should be given to the valve?

56. Describe any method of inducing a draught up the funnel. What are the advantages of induced draught over the closed stokehold system?

57. Estimate the temperature of the gases passing from the furnace when coal capable of yielding 14,000 b.t. units per pound is burnt with 19 lb. of air, supplied at a temperature of 75° Fahr., supposing half the heat of combustion to be radiated to the furnace crown and sides.

58. Explain fully the advantages of fitting vessels with closed ash-pits, and state the parts of the machinery and boilers which you consider would require special attention when forced draught is being used.

59. What increase of power would you expect to obtain from a set of boilers fitted with forced draught, compared with the same set under natural draught?

60. A set of six boilers placed back to back are required for a ship. Show by an outline sketch the general arrangement of boilers, smoke boxes, uptakes, and funnels.

61. Flame is sometimes seen near the mouth of the funnel: what causes this appearance? Is it beneficial or detrimental? Why so?

62. Distinguish between saturated and superheated steam. What percentage of water is carried in suspension by steam which has had 1100 b.t. units expended per pound in its production at a temperature of 347° Fahr. from water at 75° Fahr.?

63. When steam expands in a cylinder, describe how the pressure varies with the increase of volume. Point out the distinction between this, and the expansion of a gas by Boyle's Law, and describe the variation of temperature which a steam cylinder undergoes in the double stroke.

64. State an expression for the efficiency of a thermally perfect engine. If an engine working between the limits of temperature 400° and 100° requires an expenditure of 200 b.t. units per minute per horse-power, determine the relative efficiency.

65. What is meant by the "total heat of formation of steam"? Feed water at 62° Fahr. is pumped into a boiler and evaporated into steam containing 2 per cent of moisture at a pressure of 300 lb. (422° Fahr.). What quantity of heat is added to each 1 lb. of water in the process? What additional heat would be required to superheat the steam at 500° Fahr.?

66. Give the formula for calculating the calorific value of a fuel, and explain

how the formula is obtained. What is meant by the theoretical evaporation of a fuel in pounds of water, from and at 212° Fahr. ?

67. In a boiler 22 lb. of air are supplied for 1 lb. of fuel, and the funnel gases escape at a temperature of 800° Fahr. : find the heat wasted per 1 lb. of fuel when the temperature of the air supplied is 65° Fahr., and the specific heat of the funnel gases is 0·23°. Find the maximum efficiency under these conditions.

68. A fuel containing 87 per cent of carbon and 31 per cent of hydrogen is completely burnt in the furnace of a boiler ; the analysis of the funnel gases by weight, neglecting any steam passing away with the gases, shows 13 per cent of CO_2, 9·4 per cent of O_2, and 77·6 per cent of N_2. Calculate the composition of the funnel gases when taking the water vapour into account, and find the number of pounds of air supplied per 1 lb. of fuel.

69. Describe either a metal form of water gauge, or a plate water gauge.

70. Define " British thermal unit " and " foot-pound," and state the relation between these units. Calculate the heat, in b.t. units, equivalent to the work done in three hours by an engine developing 15 horse-power.

71. What is the boiling-point of water at atmospheric pressure, and how does the boiling-point vary with a rise of pressure ? What is the amount of latent heat, sensible heat, and total heat of formation of 1 lb. of steam at 185° Fahr., from water at 38° Fahr. ?

72. Make a longitudinal sectional elevation of a cylindrical boiler, showing the general arrangement of the various parts, and, as fully as you can, the various forms of joint adopted for the internal parts.

73. Is the steam jacket as fitted to marine cylinders of more use with the ordinary single-cylinder engine or a stage-expansion engine ? Why ?

74. Explain the terms " maximum density of steam " and " saturated steam." Calculate the " dryness fraction " of steam formed from water at 90° Fahr., at a constant absolute pressure of 225 lb. per square inch (temperature 392° Fahr.), by the expenditure of 1100 b.t. units per 1 lb.

75. What data are necessary in calculating the horse-power of a marine engine ? What is the value in " inch-tons per second " of one horse-power ? How is the horse-power of a marine steam turbine obtained ?

76. What are the chief constituents of coal which give it its value as a heat-producer ? Describe the chemical actions which take place during the combustion of coal, and state an expression for the heat value of coal.

77. Explain the formation of smoke and flame.

78. Explain the action of a chimney, or funnel, in inducing a draught. State the amount of air required per 1 lb. of coal burnt, and mention the losses which occur—first, when the supply of air is limited ; second, when the air supply is too great.

79. What are the advantages of using superheated steam, and where is the best place in which to apply the superheat ?

80. Sketch an athwartship section through a main engine bedplate of a large ship, and show how the bedplate is secured to the engine bearers.

81. The cylinders of a compound engine are 12 and 24 inches diameter, and the mean effective pressures are 70 and 20 lb. per square inch respectively ; the stroke is 9 inches, and the revolutions are 100 per minute. Find the horse-power. *Ans.* 77 I.H.P.

82. An ash-hoist engine has two cylinders of 6 inches diameter and 6 inches

stroke; the mean effective pressure in each cylinder is 60 lb., and the engine makes 2 revolutions per second: what is the horse-power? *Ans.* 12·4.

If all the horse-power indicated be utilised, at what rate will the engine raise a bucket of ashes weighing 2½ cwts.? *Ans.* 24·8 feet per second.

83. The initial steam pressure in a cylinder is 100 lb. by gauge, and cut-off is at one-third stroke: what is the average effective pressure on the piston if the steam is exhausted to the atmosphere? *Ans.* 68 lb. per square inch.

84. What is meant by using steam expansively? Show, in any way, that expansive working should be advantageous in point of economy.

85. Give an account of the action of the cylinder metal leading to waste of heat in the steam engine, and show how the compound method of expansion leads to increased economy. What fittings in connection with the cylinder tend to reduce the above waste, and in what way?

86. What is the effect of the receiver pressures in a triple expansion engine? What is the effect of making the cut-off in each of the cylinders earlier respectively? What are the means provided for doing this?

87. Explain how the condenser tube ends are made water-tight in the tube plate; and what is the result if they leak? Describe briefly the use of the air pump and the use of the feed tank.

88. The diameter of a cylinder is 30 inches, and the mean effective steam pressure on the piston is 80 lb. per square inch when developing 1600 I.H.P.: at what speed is the piston moving? *Ans.* 933·3 feet per minute.

89. Why is it better to carry out a large expansion of steam in stages, as in the three-stage expansion engine, rather than in one stage, as in the simple engine?

90. Name, and give the use of, the various materials used in the construction of a set of engines and boilers.

91. Describe the arrangement of stuffing-box now usually fitted for piston rods. Describe the various parts of which a piston for a large engine is composed.

92. With the aid of a sketch, show how a main bearing is formed. What is the object of the construction in this and other working bearings?

93. Describe the bed or sole plate of a set of vertical engines, and how the cylinders are supported. Why is it necessary to have a guide for the piston-rod head?

94. Describe the line of shafting from the engine to the propeller, naming the various bearings and their object. Why is the shafting made hollow?

95. Describe briefly, with the aid of a simple sketch, the construction of a cylinder, indicating how the steam enters and leaves.

Name the mountings usually fitted to the cylinder.

96. What is the object in fitting a steam jacket to a cylinder? Describe how the steam used in the jacket is kept distinct from that used for driving the piston.

97. Comparing the modern with the early forms of marine engines for a given power, describe briefly the reasons for—

(*a*) The increased economy of coal;

(*b*) The saving in weight and space.

98. Describe the construction of a piston suitable for a large cylinder. How is the piston secured to the piston rod?

99. Describe the arrangement of a carbon stuffing-box and gland suitable for a modern set of turbines.

100. Sketch and describe the Michell thrust bearing. Of what material are the bearing surfaces formed ?

101. Describe the construction of a piston for a large cylinder, showing how it is made steam-tight in the cylinder. Name the different materials of which the various parts are made.

102. Name and state the object of the various mountings you would expect to find on a low-pressure cylinder of a two-stage turbine.

103. Describe and explain the use of a connecting rod, giving particulars of the bearings at each end of it.

104. Name, and give the reasons for, the principal operations of the slide valve during a complete revolution of the engine.

105. Sketch a single-ported slide valve in mid position on the cylinder face, and also in the position it occupies at the commencement of the piston stroke. Mark in the sketches the *steam* and *exhaust lap* of the valve, and the *lead* of the valve. Define these quantities, and state the uses of giving lap and lead respectively.

106. Show, by means of outline sketches, the relative position of piston and crank at the four operations of an internal combustion engine.

107. What are the advantages gained by the use of the piston type of slide valve ? Describe the construction of one of these valves.

108. What is the object in fitting link motion to a set of engines ? Explain the effect of putting the link in intermediate positions between " full gear ahead " and " stop."

109. Define the terms " lap " and " lead " as applied to a slide valve, and explain their object.

110. Describe any arrangement you know of for reversing a set of turbines.

111. Explain what is meant by the " dead centre " of an engine. To what difficulty does this give rise, and how is it overcome in single and double acting engines ?

112. Give sketches showing how the steam passes through the engines in three-cylinder and in four-cylinder three-stage expansion engines. Why is the four-cylinder type used in place of the three-cylinder type in some cases ?

113. Give reasons for the superior economy obtained in modern turbines in comparison with those formerly in use. State what fuel consumption is usually obtained under good circumstances at sea (*a*) for main engines, (*b*) for all purposes at full power. Why is the consumption sometimes greater at very low powers per S.H.P. ?

114. Enumerate and explain the use of the fittings attached to a large marine turbine engine.

115. Where are reducing valves fitted on a steam cylinder, and for what reason ? Describe one, and give a sketch of its action.

116. Draw the section through the exhaust port of a low-pressure turbine, and show how the astern turbine in the same cylinder is allowed to revolve in the vacuum created by the condenser.

117. Show by sketches the method of securing a turbine in a ship, and the method of allowing for differences in expansion of cylinder and shaft. Why is the method described necessary ?

118. Sketch a thrust block of the horse-shoe type, showing how the white metal is secured to the shoe, and how the oil supply is arranged. How is the shoe kept cool when under way ? Why is cooling necessary ?

119. Describe, with sketches, how consecutive lengths of propeller shafting are connected together.

120. Why is a thrust block fitted to a marine engine ? Give a short description, aided by sketches, of (*a*) thrust block, and (*b*) thrust shaft. How is the thrust taken in a marine steam turbine ?

121. Sketch a section through the bush fitted at the inboard end of a ship's stern tube. Of what material is the bearing surface made, and how is it secured in the bush ?

122. Sketch and describe the stern-tube bearing of a large vessel. How is the stern shaft removed ? Describe the necessary fittings.

123. Sketch and describe the ordinary form of piston rod and crosshead, showing how the rod is attached to the piston, and the provision made for lining up the shoe. What is the use of a stretching length, and where are they fitted ?

124. Describe, with sketches, the labyrinth packing used for a high-pressure turbine, distinguishing between the axial and radial types.

125. Sketch the usual arrangement adopted for independently linking up engines, showing how the gear is made practically inoperative when going astern.

126. Show by sketches the positions of a simple slide valve on the valve face at the points of admission, cut-off, release, and compression. Indicate the direction of rotation obtained by the motion of the piston.

127. Define "open" and "crossed" rods as applied to link motion. What is the effect in each case on the "lead" when "linked up" ? How is the "angular" advance obtained, and why is it used ?

128. Describe the principal differences between a jet condenser and a surface condenser. Which type is now used, and why ?

129. Describe a form of air pump worked by an independent engine. Sketch and describe an air pump with only one set of valves. Sketch a Kinghorn metallic valve and seating, and describe its action.

130. What is meant by the "closed exhaust" system ? Explain any reason for its retention in modern vessels.

131. Describe the arrangements made for draining the steam and exhaust systems. Sketch and describe an expansion steam drain trap, and state your reasons for its action.

132. Describe the fittings and pipes attached to a feed tank. Give an outline sketch.

133. Sketch, in plan, an outline arrangement of main steam pipes of any modern twin-screw ship with which you are acquainted, indicating the position of all valves and expansion joints.

134. Sketch a flange joint of a steel main steam pipe, showing how the flange is secured to the pipe. Describe some methods of making the joint between the flanges, and state your opinion of the advantages of any particular type of joint.

135. Sketch a modern type of evaporator suitable for working with exhaust or low-pressure steam, showing as many of the mountings as you can.

136. Describe a method of keeping a constant density in an evaporator. What precautions are necessary to keep the apparatus in working order ?

137. Describe the method of supporting and staying the cylinders of a large internal combustion engine. How is expansion allowed for ?

138. Name the parts passed through by the condensed steam from the air pump to the feed tank, and state the means adopted for minimising the water loss.

139. Describe the working of an evaporator, and state the various pressures in its parts when utilising steam of low pressure. What is the advantage of generating steam at low absolute pressures in evaporators, and what precautions should be taken to prevent priming ?

140. Name some of the various filtering materials which are used for the purification of boiler feed-water on board ship. Why is a filter necessary ? Sketch and describe any form of filter with which you are acquainted.

141. Sketch a section through the barrel of a feed pump, showing the suction valves and delivery valves, and the method of packing the water piston.

142. Sketch and describe a feed-water heater. What are the advantages of feed heating ?

143. A vessel can steam 12 knots on a fuel consumption of 45 tons per day ; the bunker capacity is 375 tons. What would be the greatest speed at which she could steam between two ports 1536 sea-miles apart ? *Ans.* 15 knots.

144. What are the principal resistances offered to the passage of a vessel through the water ? Show how the relation " I.H.P. varies as the speed cubed " for moderate speeds is obtained.

145. What is the object in making provision for the independent adjustment of the cut-off in a particular cylinder of a set of stage-expansion engines ?

146. With the aid of a diagram, indicate what is meant by the " angular advance " of the eccentric, and explain why this is given.

147. What is the object in fitting a double-ported valve in preference to a single-ported slide valve ?

What arrangements are made for reducing the friction between the valve and cylinder face as much as possible ? Describe a balanced relief slide valve.

148. Why are piston slide valves fitted to some cylinders and flat slide valves fitted to others ?

Describe the relief arrangements fitted to a large flat slide valve.

149. How are the tubes of a condenser secured in place ? Explain the object aimed at in the method you describe.

150. Explain the object and method of action of an air pump for a surface condenser, and state how the pump is worked in a turbine plant.

151. Sketch and describe an ejector air pump. What advantage is obtained by this arrangement ? Of what materials is the pump made ?

152. Describe a circulating pump and its action. Make an outline sketch of the course of the cooling water through a surface condenser, and describe how you would pump out the engine-room bilge with a circulating pump.

153. About how much cooling water is required per S.H.P. for an ordinary turbine engine ? And why ?

154. Describe the general arrangement of condensers, air pumps, and circulating pumps as fitted in a large modern ship.

155. Name the fittings generally to be found on a condenser, and state their uses.

156. Give the names and describe the object of the various valves through which the steam has to pass on its way from the boilers to the engines. Why is it necessary to have a " nozzle control " valve as well as the regulating valve fitted at the engines ?

157. What is the use of an " emergency valve " ? Describe one, and state from where it can be worked.

158. Describe the use and position of a boiler steam dryer.

159. By means of a sketch, show the arrangement of main steam pipes and valves in the engine room of a ship, giving the object and name of the different valves and fittings.

160. Describe the construction and action of a steam regulating valve, and state the arrangements made for effecting small alterations in the speed of the engine.

161. Describe, with the aid of a simple sketch, the arrangement of feed pipes and pumps you would expect to find in a large vessel.

162. Describe the fittings to be found in connection with a feed tank. Why is a filter tank fitted in some vessels as well as a feed tank ?

163. Sketch and describe a feed-water filter or grease extractor. Where is it fitted ?

164. Why is it necessary to use fresh water only with modern boilers ? What means are adopted to reduce the waste of fresh water in the machinery department of a vessel to a minimum ?

165. Describe the method of " making up " the feed-water loss.

166. How is the condition of the feed water for boilers, as regards freshness, ascertained on board ship ? To what causes would you ascribe a gradual rise in density ?

167. Describe, with the aid of a sketch, the course of the feed water from the condenser to the feed pump.

168. Describe an evaporator, and state its use. What is meant by the terms (*a*) generating steam, (*b*) gained steam, (*c*) coil drain, (*d*) vapour valve, as applied to evaporators and fittings ?

169. What is a distilling condenser used for ? Describe the method of working one, and the manner in which cold water can be produced.

170. If 900 I.H.P. is sufficient to give a Fleet Scout a speed of 12 knots, what power is required for a speed of 15 knots ? *Ans.* 1758 I.H.P.

171. Two screws, each of 10·5 feet pitch, drive a vessel at $21\frac{3}{4}$ knots, when the engines make 250 revolutions per minute : what is the slip per cent of each propeller ? *Ans.* 16.

172. Describe briefly the apparatus used for maintaining a supply of fresh water for boilers and drinking purposes on board ship.

173. Name and briefly state the object of the various auxiliary engines usually fitted in a battleship. Which of them are usually fitted in the boiler rooms ?

174. Enumerate all the pumps you can think of which are fitted in a large ship, and state their uses.

175. Describe the course of the steam after leaving the boiler stop valve to its return as water to the feed tanks in a three-stage expansion turbine engine.

176. Describe the features of the modern screw propeller. Of what material is it made, and why ?

177. Describe the construction and object of an eccentric. Show by means of a simple diagram the position of the eccentrics on the shafting to produce ahead or astern motion.

178. If a ship burns 200 tons in steaming 800 miles at 11 knots speed, how much will she require for a distance of 1200 miles at 10 knots, excluding coal required for auxiliary purposes in each case ? *Ans.* 248.

179. Describe fully a method of determining the pitch of a screw propeller.

180. In a certain vessel a speed of 30·4 knots was obtained with 387·0 revolutions per minute; the pitch of the propellers is 9 feet 4 inches. What was the slip per cent? What revolutions would be required to give a speed of 32 knots? *Ans.* 15 per cent; 408 revolutions.

181. Describe briefly what is meant by "cavitation." How has its effect been reduced? And what causes tend to limit the increase in blade area of naval screw propellers?

182. A ship, developing 3000 I.H.P., steams 12 knots on a coal consumption of 2·24 lb. per I.H.P. per hour. What is the shortest time in which she could make a port 900 miles distant with 400 tons of coal available? What would be the corresponding approximate coal consumption per I.H.P. per hour, and the approximate I.H.P. developed?

183. Explain clearly what is meant by the "clearance volume" of a cylinder, with special reference to an oil engine.

184. How is the total ratio of expansion affected by clearance? Define the "ratio of expansion" as applied to a state-expansion turbine.

185. Construct as accurately as you can the elementary indicator diagram of an engine working under the following conditions:

(*a*) Steam pressure at the engines, 129 lb. per square inch by gauge;
(*b*) Cut-off at one-third stroke;
(*c*) Absolute back pressure (mean), 10 lb. per square inch.

186. Describe an indicator, showing how the vertical travel of the pencil is assured. What precautions are required to obtain a correct indication of the pressure in cylinders?

187. Describe, with sketches, the hunting gear of a steering engine. Describe the gearing connecting the steering engine with the rudder head. How is screw gearing made self-holding, and why is it necessary to allow a certain amount of fore-and-aft play of the shaft in its bearings?

188. Describe Brown's telemotor controlling gearing. Sketch any form of joint which you would think suitable for the hydraulic piping.

189. Explain the principle on which ice is made by an ice machine of the CO_2 type. Sketch any system of producing ice with which you are acquainted.

190. Describe, with sketches, the best method you know for lubricating the crank bearings of a large engine. How would you lubricate the other principal bearings? Describe the syphon lubricator, and give the necessary precautions for adjusting its proper working.

191. Sketch and describe a valveless pump such as is commonly employed for forced lubrication. Describe the arrangement for lubricating each principal bearing.

192. A twin-screw ship, having two main groups of boilers, is under way with one group of boilers alight; orders are given to raise steam for full speed. Explain how you would proceed to raise steam in, and connect up, the other group of boilers, and mention the precautions you would take.

193. Describe the nature of the examinations you would make before taking over the duties of Officer of the Watch in the engine room of a modern large vessel under steam, with water-tube boilers.

194. Describe fully the examinations you would make of a set of marine engines after a long voyage under steam.

195. What precaution is particularly necessary before opening out the steam valve of a water-tube boiler?

196. Describe a form of reversing gear, and the process of warming engines and getting under way.

197. Name the various kinds of water-tube boiler of which you have any knowledge, and divide them into classes—one with nearly vertical tubes and the other with nearly horizontal tubes. Describe any one example.

198. How is the independent expansion of the tubes allowed for in the following boilers respectively ?—Babcock, Niclausse, Yarrow, Thornycroft, and White-Forster.

199. Describe any method of atomising liquid fuel with which you are acquainted. What advantage is gained by heating the air supplied for combustion ? How is the water removed from the oil ? Why is it necessary to employ brickwork ?

200. Describe, with sketches, a Parsons reaction turbine. Explain how the thrust of the propeller is taken, how the main bearings are fitted and lubricated, and how the shaft is packed where it leaves the turbine cylinder.

201. What arrangements are made to produce a better economy at low powers in a turbine-propelled vessel ? Why is the turbine generally less economical at low powers than a reciprocating engine ?

202. At what speed of periphery would you expect a turbine to revolve when economy of fuel is obtained ? And why ?

203. Sketch and describe a method of attaching the blades in a turbine engine. Explain how you would determine the horse-power of a turbine engine when used (*a*) for propulsion of a vessel, (*b*) for propulsion of an electric generator.

204. Sketch and describe a method of producing astern motion of turbine-propelled vessels.

205. Describe a method of aligning the piston, rod, and guide.

206. How would you test the alignment of the crank-shaft and propeller-shaft bearings ?

207. Describe the method of fitting and adjusting a crank-head bearing. How much clearance would you allow for a 16-inch pin (*a*) at the crown, (*b*) at the horns ?

208. When the engines are suddenly eased down, how would you check the generation of steam in your ship ?—naming the type of boiler.

209. Describe how you would test the feed water for acidity, and for quantity or proportion of sea-water mixture.

210. Show, by sketches, the method of plugging a damaged tube in an upright water-tube boiler.

211. Describe the system of firing when burning North Country and Welsh coals respectively. How can you prevent flaming and decrease smoke ?

212. How would you find the steam consumption per hour from an indicator diagram ? From which cylinder would you expect the most accurate result ?

213. What effect has "linking up" the L.P. valve gear on (*a*) the economy of steam, (*b*) the relative power developed in the several cylinders, (*c*) the total power of the engine ? Why ?

214. Describe how you patch a corrugated furnace. If the original thickness is $\frac{3}{4}$-inch, what should be the thickness of the patch ? And why ?

215. A funnel is 90 feet high above the fire-grate : about how much coal per square foot of grate can be burnt without using mechanical draught ?

216. Explain a method of analysing the funnel gases and recording the

proportion of CO_2 present in them. If the proportion is found to be 20 per cent, what would you do ?

217. In using superheated steam, where would you look for possible damage to the superheater and to the machinery ?

218. Describe the method of fitting and adjusting new restricted packing rings into a piston and cylinder, and state the clearances you would allow for bronze rings working on steel liners.

219. If a steam loop appears in the H.P. indicator diagram, what would you do ? How does a steam loop affect (*a*) the practical working of the engine, (*b*) its thermal efficiency, (*c*) its mechanical efficiency ?

220. Describe a method of measuring the shaft horse-power of a large marine engine, and state the necessary particulars for its determination.

221. Show, by sketches, the best method you know of cutting oil grooves in a crank bearing. What is an oil-containing strip ?

222. Explain why a high vacuum produces better economy in a turbine than in a reciprocating engine. Sketch one method of increasing the vacuum in turbine practice.

223. Explain the system of ignition employed in (*a*) Gas-oil engine, (*b*) Diesel oil engine. In a Diesel engine (four-stroke cycle) the mean pressure is 48 lb. per square inch for each of four cylinders, diameter 20 inches, length of stroke 50 inches, revolutions per minute 240 ; estimate the I.H.P.

224. Describe the process of opening, clearing, closing up, and testing a water-tube boiler as fitted in your ship or in a steam pinnace. State any particular precautions you would take during the process.

225. As Engineer Officer of the Watch, what would you do (*a*) if a bearing becomes heated, (*b*) boiler runs short of water, (*c*) feed pump refuses duty ? If the Senior Engineer comes below and gives you an order about any of these matters, who is Officer of the Watch ?

226. Sketch and describe an oil fuel sprayer of the mechanical pressure type, and explain its action in promoting atomisation, vaporisation, and good combustion. If the supply of fuel oils fails, what substitutes can be used for (*a*) boiler firing, (*b*) internal combustion engines ? Describe any process of which you have knowledge for obtaining combustible fuel from carbonaceous substances or potatoes.

227. Explain what is meant by flash-point, burning-point, and range of explosion of combustible mixtures. How is the fuel ignited in various types of internal combustion engines ? Why is it necessary in practice to limit the compression in petrol engines ? Describe the construction of an electrically heated Hot-Bulb ignition system, and give reasons for any advantages it may have over the dual system of petrol and paraffin carburettors.

228. Describe the opposed piston type of internal combustion engine, and give reasons for the better balancing expected as compared with the ordinary single piston type. State reasons for any other comparative advantages.

229. Give a brief explanation of the Still thermal cycle of operations and of its possible advantages in relation to fuel economy. Why is it practicable to make the thickness of cylinder less compared with an ordinary Diesel type engine ? Explain fully the reason for the practical limitation of diameter of cylinder in the latter type of engine.

230. Sketch and describe a convergent-divergent nozzle used for steam turbines, and give reasons for the various details of its construction and shape.

Show by an outline curve the relation between the pressure, velocity, and temperature in an impulse turbine of the Rateau type, and explain how the various transformations of energy take place in the respective stages of expansion.

231. Sketch and describe an arrangement for measuring the propeller thrust of a marine turbine by using the Michell type thrust for the purpose.

232. Sketch and describe the method of attaching the blades to the rotor of a turbine, and explain why particular care is necessary to make the attachment secure against centrifugal action of blades moving at high peripheral velocities.

233. Why is it necessary to fit large stays between the cylinder heads and bottom framing of Diesel type engines ? Sketch and describe how this is effected in any engine with which you are acquainted.

234. Explain the difference in the compression of air when conducted isothermally and adiabatically, illustrating your reply by reference to the compression of air in a torpedo air compressor, and during the compression stroke of a Diesel engine.

235. Describe the four-stroke and the two-stroke cycles used in gas engines, and give a diagrammatic section through the cylinder of a gas engine of each type, lettering and indexing the various ports and valves.

236. Describe a system of governing internal combustion engines.

A four-stroke-cycle engine shows a mean effective pressure calculated from the indicator diagrams of 45 lb. per square inch, but misses explosion once in every 20 revolutions ; if the stroke of engine = 20 inches, diameter of cylinder = 8 inches, mean revolutions per minute = 240, estimate the I.H.P. per hour.

237. Give an account of—

(1) The pressure system of injecting oil fuel into an internal combustion cylinder ;

(2) And briefly of the nature, properties, and source of supply of a fuel suitable for propulsion.

238. What is meant by efficiency as applied to marine propulsion ? Compare the efficiencies of a modern geared turbine and a heavy oil engine in screw-propelled vessels.

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